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Applied Modal Analysis of Wind Turbine Blades

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Abstract

In this project modal analysis has been used to determine the natural frequencies, damping and the mode shapes for wind turbine blades.

Different methods to measure the position and adjust the direction of the measuring points are discussed. Different equipment for mounting the accelerometers are investigated and the most suitable are chosen.

Different excitation techniques are tried during experimental campaigns. After a discussion the pendulum hammer were chosen, and a new improved hammer was manufactured.

Some measurement errors are investigated. The ability to repeat the measured results is investigated by repeated measurement on the same wind turbine blade. Furthermore the flexibility of the test set-up is investigated, by use of accelerometers mounted on the flexible adapter plate during the measurement campaign. One experimental campaign investigated the results obtained from a loaded and unloaded wind turbine blade. During this campaign the modal analysis are performed on a blade mounted in a horizontal and a vertical position respectively.

Finally the results obtained from modal analysis carried out on a wind turbine blade are compared with results obtained from the Stig Øyes blade_EV1 program.

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1 Introduction

Wind turbines with a rotor diameter exceeding 2 m must have a type approval in accordance with the Danish approval system. As a part of the certification procedure for wind turbine blades it is recommended to determine the natural frequencies and damping of the blade. Normally these dynamic characteristics are determined for the first and second flapwise and first edgewise natural frequencies. Furthermore the first torsion frequency is determined.

In the recent years the wind turbine blades are getting larger and relatively more flexible than earlier. This has caused increasing attention to instability problems due to this matter. A well-known example of a instability problem that might lead to global failure of the whole mill is the edgewise vibrations.

In connection with development of new wind turbine blades in the future there is a need for verification of the structural properties (mass- stiffness- and damping) in a better way than it is possible today. That will give one the possibility to adjust these properties already on the proto type stage. By this it is possible to reduce the probability for instability of the blade.

A verification of the dynamic behavior of the structure can be done by extension of the traditional dynamic characterization of the blade with a determination of the mode shapes. The present report deals with a test procedure that provides such information.

Modal analysis is a process of determining the modal parameters for a construction and it is the common used method to characterize the dynamic properties of a mechanical system. It can be accomplished through experimental techniques and produces interpretable results.

The present modal analysis is performed as a system analysis where there is an input and an output signal. Because the wind turbine blade is a large structure it is necessary to treat the blade in cross sections successively. The experimental procedure is based on impact modal testing where exciting of the blade is performed by a hammer impact, at a fixed point during the test. At every cross section the acceleration are measured by three accelerometers, two in flapwise direction and one in edgewise direction, in order to determine the flap- and edgewise properties of the blade.

The present report is performed in continuation of the report “ Modal Analysis of Wind Turbine Blades” Risø-R-1181(EN), see [1]. At this project there were obtained good results doing modal analysis on a 19.1 metre wind turbine blade. It was demonstrated that it was possible to determine the natural frequencies, damping and the mode shapes by use of modal analysis. In the project the excitation were done by a hammer. It was stated that it was difficult to repeat the direction and level for the input force. Furthermore it was stated that there were a good agreement between the results obtained by experimental work and results obtained by a FEM-analysis.

The present experimental work aims at develop the method and the equipment such as it is useable in practice. Furthermore there is focus on systematic errors such as elasticity of the test rig, uncertainty of measuring the points for mount-

ing of the accelerometers and the ability to repeat the test without variations in the measured results.

2 Blade reference co-ordinate system

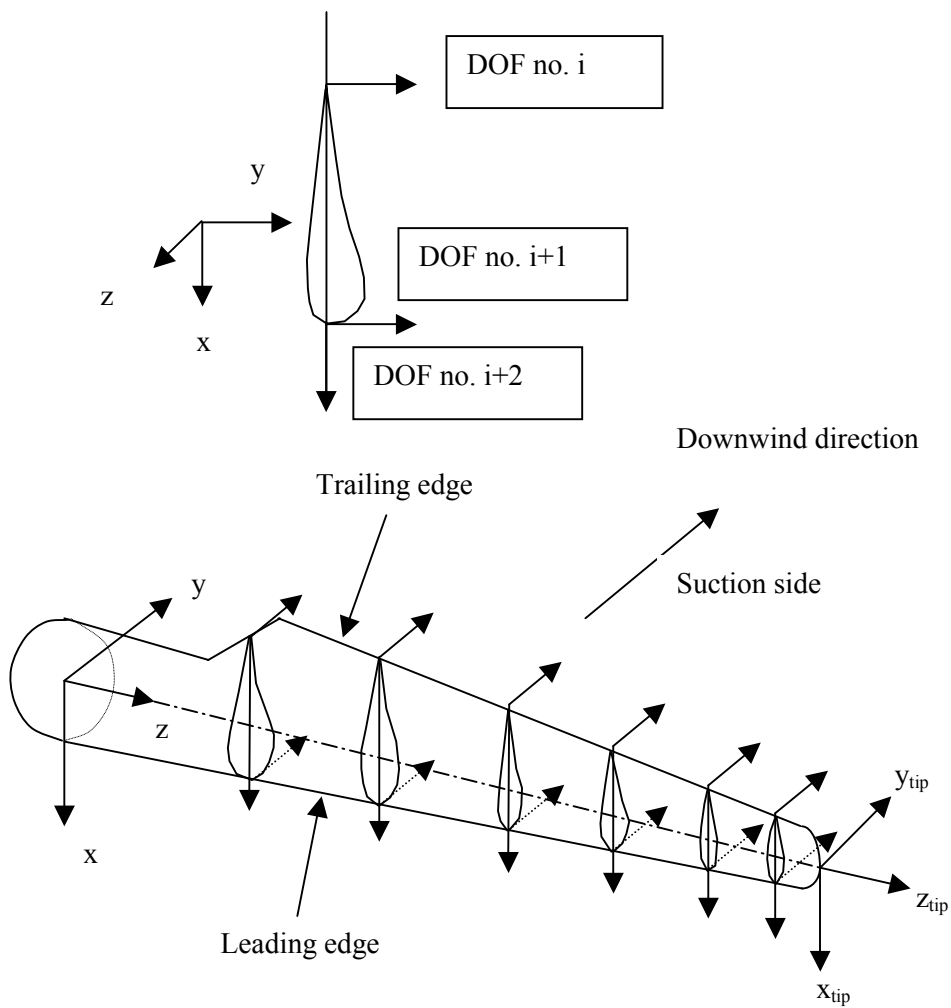


Figure 1. Blade reference co-ordinate system. The upper sketch in Figure 1 shows a cross section of the blade. The lower shows the global co-ordinate system of the blade in modal analysis.

3 Definitions

In this chapter there is a short introduction to the theory of modal analysis used in the experiments with wind turbine blades.

Modal analysis is the process of determining the modal parameters, which are then sufficient for formulating a mathematical dynamic model. The free dynamic response of the wind turbine blade can be reduced to a discrete set of modes. The modal parameters are the natural frequency, damping and mode shape of the wind turbine blade. Within the frequency range of interest the modal parameters of all the modes constitute a complete dynamic description of the wind turbine blade. The modes of vibration represent the inherent dynamic properties of the wind turbine blade. The range of applications for modal data includes, checking the modal parameters, verifying and improving analytical models, predict the response to assumed excitations, predict the change in dynamic properties due to physical modifications, i.e. load or stiffness, predict the necessary physical modifications required to obtain a desired dynamic property.

In the experiments it is not possible to measure the motion in all points on the structure. It is necessary to treat the structure in cross sections successively. It is possible to use a finite number of degrees of freedom to describe the blade motion. The mode shapes of the blade are assumed to be described by deflection in flap- and edgewise direction. Torsion is assumed to be described by rotation of the chord about the pitch axis. The rigid body motion can be described by three DOF's (degree of freedom) in each cross-section. Every DOF have a position and a direction on the structure, which are used to the description of the rigid body motion. Two flapwise DOF's describe the flapwise deflection and torsion, and one edgewise DOF describe the edgewise deflection.

The rigid body motion can be derived as a function of the three amplitudes of the DOF's in the following forms:

$$U = Ax$$

where U is the motion of the cross section and x is the corresponding amplitudes in the three DOF's of the cross-section.

$$U = \begin{Bmatrix} u_x \\ u_y \\ \theta \end{Bmatrix} \quad \text{and} \quad x = \begin{Bmatrix} x_i \\ x_{i+1} \\ x_{i+2} \end{Bmatrix}$$

A is a three by three matrix given by the positions of the three DOF's.

When using this relation a mode shape of the blade can be estimated in a number of cross-sections. The number depends on how large the structure is.

Each cross section is assumed to undergo a rigid body motion, in a plane perpendicular to the pitch axis. In each plane there is mounted three accelerometers. Two accelerometers on the leading edge measures the deflection in edgewise and flapwise direction respectively. One accelerometer on the trailing edge measures the deflection in flapwise direction. It is assumed that the edgewise deflection is equal at the leading edge and at the trailing edge.

We assume that the system have a linear behavior so that the response is proportional to the excitation.

The most common type of modal testing uses a FFT analyzer to measure a set of Frequency Response Functions (FRF's) from a structure and then use a parameter estimation (curve fitting) method to determine the structures modal properties from the FRF measurements. The curve fitting method typically fits an analytical model to the FRF data and this process determines estimates of the unknown modal parameters of the model. The common used fitting method is the Rational Fraction Polynomial Method (RFP) [4]. These parameter estimates are then assumed to be the correct modal parameters of the structure.

Time domain model

The modal properties can be considered as an eigen value problem. The natural frequencies and logarithmic decrements are the eigen values, and the mode shapes are the eigenvectors. The linear equation of motion with external excitation can be written:

$$[M]\{\ddot{x}(t)\} + [C]\{\dot{x}(t)\} + [K]\{x(t)\} = \{f(t)\}$$

where,

$[M]$ =(n x n) mass coefficient matrix

$[C]$ =(n x n) damping coefficient matrix

$[K]$ =(n x n) stiffness coefficient matrix

$\{\ddot{x}(t)\}$ = n-dimensional acceleration vector

$\{\dot{x}(t)\}$ =n-dimensional velocity vector

$\{x(t)\}$ = n-dimensional displacement vector

$\{f(t)\}$ =n-dimensional external force vector

The modal properties used in the following experiments are based on a formulation of the blade dynamics in the frequency domain [5].

Frequency domain model

The dynamics of a structure can equivalently be described in the frequency domain by a transfer function. A transfer function matrix model describes the dynamics between n-DOF's of the structure and contains transfer functions between all combinations of DOF pairs.

This linear model can be written in terms of FRF's as:

$$\{X(\omega)\} = [H(\omega)]\{F(\omega)\}$$

where:

$\{X(\omega)\}$ =n-vector of Fourier transformed displacement responses

$\{F(\omega)\}$ =n-vector of Fourier transformed force input

$[H(\omega)]$ =(n x n) matrix of FRF's

n=number of test degree of freedom (DOF's) on the structure

ω =The frequency variable

The FRF matrix can be written in terms of modal parameters as:

$$[H(\omega)] = \sum_{k=1}^{Modes} [R_k] / 2j(j\omega - p_k) - [R_k] / 2j(j\omega - p_k^*)$$

where:

$[R_k]$ = (n x n) matrix residues for mode (k)

$p_k = \sigma_k + j\omega_k$ = complex pole location for mode (k)

σ_k = modal damping for mode (k)

ω_k = modal frequency for mode (k)

modes = the number of modes in the model

The relationship between natural frequencies (f_k), logarithmic decrements (δ_k) and the eigen values are:

$$f_k = 2\pi\omega_k$$

$$\delta_k = -\sigma_k / f_k$$

The term damping factor (σ_k) are sometimes used instead of logarithmic decrement (δ_k) as a measure for damping.

It can be shown that the mode shape can be obtained from a row or column of the residue matrix $[R_k]$ for each mode (k) since the residues are related to the mode shape by the formula:

$$[R_k] = A_k \{U_k\} \{U_k\}^t \quad k=1, \dots, \text{modes}$$

where:

$\{U_k\}$ = the mode shape for mode (k), an n-vector

A_k = a scaling constant for mode (k)

t denotes the transpose of the mode shape

At least one row or column of FRF measurements are typically made and these measurements are curve fit to obtain the modal pole locations (frequency and damping). A row of transfer functions can be obtained from an experiment by measuring the response in all DOF's, while the point of excitation is fixed to one DOF. To obtain a column of H the response is measured in one DOF while the point of excitation is moved between all DOF's. Either one of these procedures can be used to obtain all mode shapes. Each mode is represented in an FRF by two complex parameters, a complex pole location and a complex residue.

See also [2] and [3].

Accuracy

The accuracy of the measured natural frequencies, damping and mode shapes depend on three things. First the accuracy of the DOF characteristics is depended on how accurate the positions and directions of the accelerometers are determined. Second a non-linearity of the blade dynamics will distort the transfer functions yielding that the theoretical basis for the curve fits become invalid.

Third the measurement errors on the transfer functions influence the curve fits, which are used to estimate the modal properties. Measurement errors can have many sources. Some of them can be false calibration, vibrations from other machines during measurements, and a flexible test rig [1].

4 Discretization of the blade

Wind turbine blades are large structures. The aim of the modal analysis is to determine the modal parameters for a number of mode shapes. It demands a number of DOF's to give sufficient data to determine the mode shapes of the blade. To have sufficient number of DOF's the blade is discretized into a number of cross sections. Each cross section has three DOF's, one in the edgewise direction and two in flapwise direction.

4.1 Determination of measurement-points and co-ordinates

Description of test

In general it is necessary to have a number of cross sections that is at least at the same order as the order of natural frequency of interest. In other words, if one is interested in the fourth flapwise natural frequency four cross-sections are necessary. Ideally one should position the DOF's where the biggest amplitudes are. Figure 2, a graph of a blade shows a node in 50% of blade length and maximum amplitude in app. 70% of blade length. As the next figure (Figure 3) shows nodes and maximum amplitudes from different mode shapes might coincident. One disadvantage by having a DOF where the structure has a node is a bad signal-to-noise ratio and by that a risk of inaccuracy on the mode shape.

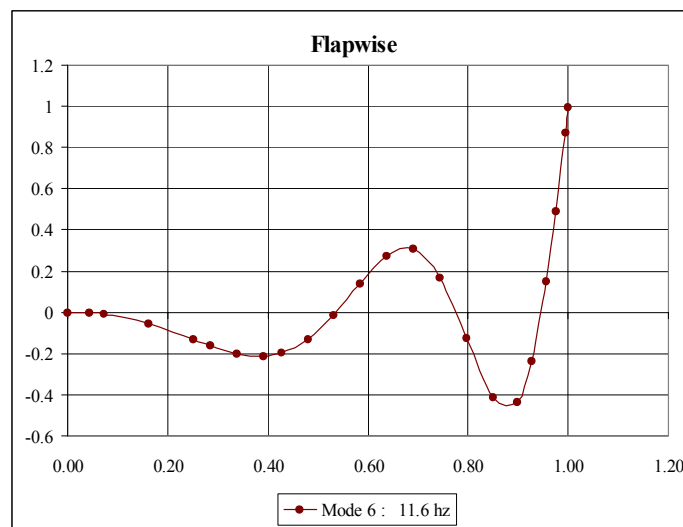


Figure 2. Mode shapes of a blade based on Stig Øyes Blade_EV1 program

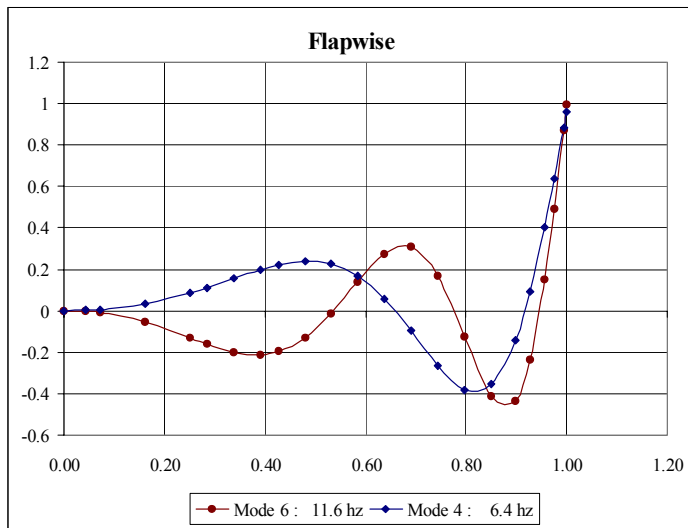


Figure 3. Mode shapes of a blade based on Stig Øyes Blade_EV1 program

As the blade is discretized, measurement-points are established on both leading- and trailing edge of the blade for each cross-section of the blade. The coordinate of each of the measurement points are determined in a coordinate system with the pitch axis at the blade root interface as origin. The measurements are carried out to create a model in the modal software and to determine the mode shapes of the structure.

In order to determine the need for accuracy when positioning and measuring the measurements-points in modal analysis calculations have been made. The aim was to determine the size of error when the measurements were carried out with two different incorrect angles during determination of coordinates.

The demands for accuracy influences the time consumption and the work carried out prior to the test.

Determining DOF positions

In the sketch in Figure 4 it is outlined how the determination of the coordinates are done. The coordinate system has its origin in the centerline of the test rig.

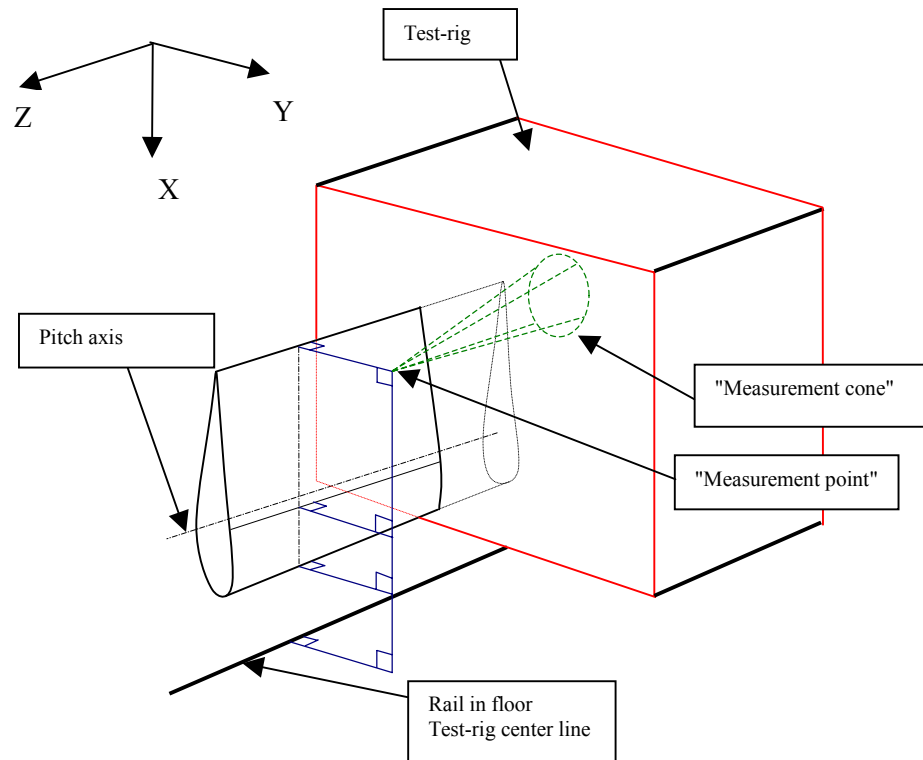


Figure 4. Sketch of coordinate system and “measurement-lines” used for determination of accelerometer positions

Measurements in Z-direction

The first coordinate determined is the positions in the Z-direction. The Z-direction is the longitudinal axis of the blade. The positions in this direction is chosen either by the designer of the blade based on a model used for calculations during design, or positions are chosen by those who carries out the test. The positions are chosen with a closer distance between the cross-sections in the tip-end of the blade than in the root-end, while the change in deflection per length unit is higher in the tip-end than in the root-end.

The Z-positions of the accelerometers are determined by measuring the horizontal line from the accelerometer position to the test-rig cf. Figure 4. Possible faults on the Z-distance are measuring off parallel to the centerline, as sketched as a cone on Figure 4.

By an angular error of 5° or 10° the following errors are seen on the coordinates in Z-direction.

Table 1. Errors on measurements in Z-direction.

Distance from Root interface [m]	5° error	Radius of measurement cone [m]	10° error	Radius of measurement cone [m]
13	13.05	1.13	13.25	2.26
18	18.07	1.57	18.35	3.13
23	23.09	2.00	23.44	3.99
26	26.10	2.27	26.50	4.51
28	28.11	2.44	28.54	4.86
30	30.11	2.61	30.58	5.21
32	32.12	2.79	32.62	5.56
34	34.13	2.96	34.66	5.90

Column one shows the distance from one measurement point to the root interface. Column two shows the measured distance. Column three shows the offset from parallel, (or radius of cone on Figure 4). Column four and five shows the value for 10° error.

The error is app. 0.4 % for the 5° error and 2% for the 10° error.

Measurements in Y-direction

The second coordinate determined is the positions in the Y-direction. The Y-direction is in the direction perpendicular to the chord of the blade. The DOF's in the Y-direction are positioned on both the trailing edge and the leading edge. The active direction of the accelerometers in the Y-direction is the flapwise direction of the blade.

The Y-positions of the trailing edge accelerometers are determined by measuring from the trailing edge to a thought vertical line that is off the blade surface. On the floor the distance from the vertical line to the rail in the floor is determined. This gives the coordinate in the trailing edge Y-direction. On the leading edge the offset off the vertical line from the floor rail is determined.

Possible faults on the Y-distance are measuring off the right angles shown on Figure 4.

Table 2. *Errors on measurements in Y-direction.*

By measurement in the y-direction there are the following errors on the measurements.		
Distance to vertical [m]	5° error	10° error
0.05	0.0502	0.0508
0.10	0.1004	0.1015
0.20	0.2008	0.2031
0.30	0.3011	0.3046
0.40	0.4015	0.4062
0.50	0.5019	0.5077
1.00	1.0038	1.0154
1.20	1.2046	1.2185

The error is app. 0.4 % for the 5° error and 2% for the 10° error.

Measurements in X-direction

The last coordinate determined is the positions in the X-direction. The X-direction is in the direction of the chord of the blade. The DOF's in the X-direction are positioned on either the trailing edge or the leading edge. The active direction of the accelerometers in the X-direction are the edgewise direction of the blade.

The X-positions of the leading edge accelerometers are determined by measuring from a vertical line based in the centerline to the floor and subtract the measure from the X-accelerator position to the floor. The position of the trailing edge X-position is measured as the vertical line on the sketch in Figure 4, subtracted from the centerline measure.

Possible faults on the X-distance are measuring off the vertical line between accelerometer positions and the floor cf. Figure 4.

Table 3. Errors on measurements in X-direction.

By a test rig centerline height of X meters the error on x-coordinate will be of the following magnitude.				
Distance to floor X [m]	5° error	Distance off vertical [m]	10° error	Distance off vertical [m]
1.00	1.0038	0.0872	1.02	0.1736
1.25	1.2548	0.1089	1.27	0.2171
1.50	1.5057	0.1307	1.52	0.2605
1.75	1.7567	0.1525	1.78	0.3039
2.00	2.0076	0.1743	2.03	0.3473
2.50	2.5095	0.2179	2.54	0.4341
3.00	3.0115	0.2615	3.05	0.5209
3.50	3.5134	0.3050	3.55	0.6078

Conclusion

Generally the 5° error gives a variation in the determined distances on app. 0.4 %. By these measurements in the Z-direction, and if a 5° error is acceptable, the circle on the test rig, in which the measurement device hits, has a diameter of app 6 meters when the measured distance is 35 meters. When the measurement is carried out within a circle of 1 m in diameter on the test rig, the magnitude of the error is app. 0.7° over a distance of 40 meters. In practice it is possible to determine the distances within this 0.7° error.

4.2 Angular error on accelerometer measurements

As the position of the accelerometers are of importance the angular direction of each accelerometer is of the same importance. The positioning of the accelerometer mounting plates is a time requiring job that demands high accuracy.

Figure 5 shows the mounting plate for the accelerometer on a trailing edge DOF. Two plates have been mounted and the top fixture is mounted 5 ° off the vertical plane.



Figure 5. Angular steel plates attached to the trailing edge, mounted with accelerometers with magnetic footplates.

Description of test

The magnitude of error when the accelerometers are positioned out of the same plane or out of one plane perpendicular to another is of same size as for the co-ordinate determination of accelerometer positions, i.e. in the magnitude of 0.4% with an error of 5° .

To determine the influence of an angular error on the modal parameters two tests were carried out. The first test was a test with no angular error on the accelerometers, the second test was carried out with a 5° error on the trailing edge accelerometers. During the second test there was no error on the leading edge accelerometers. The measurements on the trailing edge are used to determine both flap- and torsional parameters.

The angular error is expected to influence on the mode shapes, i.e. causing that the accelerometers detect a smaller deflection of the blade in the test where the accelerometers are angled out of the correct plane. Though the angular error has little influence on the determination of coordinates it is expected that a 5° error have very little influence on the mode shape

The smaller deflection is expected while the angled accelerometer is measuring only a component of the actual acceleration.

Experimental test set-up

The blade used for the test was a Nedwind 25 blade, a 12-meter blade.

The tested blade was mounted in the test rig, with the trailing edge upwards.

The blade was discretized in 6 cross-sections, i.e. 18 DOF's.

Cross-Section	DOF	Z [m]	X [m]	Y [m]	Active direction
1	1	1.97	-0.95	0.08	Y
1	3	1.97	0.35	-0.17	X
1	2	1.97	0.35	-0.17	Y
2	4	4.00	-0.90	0.03	Y
2	6	4.00	0.36	-0.15	X
2	5	4.00	0.36	-0.15	Y
3	7	6.00	-0.78	-0.03	Y
3	9	6.00	0.31	-0.16	X
3	8	6.00	0.31	-0.16	Y
4	10	8.01	-0.66	-0.09	Y
4	12	8.01	0.25	-0.17	X
4	11	8.01	0.25	-0.17	Y
5	13	9.99	-0.54	-0.15	Y
5	15	9.99	0.19	-0.19	X
5	14	9.99	0.19	-0.19	Y
6	16	11.79	-0.36	-0.19	Y
6	18	11.79	0.14	-0.19	X
6	17	11.79	0.14	-0.19	Y

The blade was excited in both edgewise and flapwise direction simultaneously by impact of the modal hammer mounted with the force transducer. The hammer with force transducer was set up to excite the blade in a distance of 7 meters from the root intersection under an angle of 45° to both the flapwise and edgewise direction. The point of attack was on the suction side of the leading edge. The force direction was perpendicular to the Z-direction of the blade.

Modal test

The test was carried out in three phases with measurements in two cross-sections in each phase. The measurements were made from the root end of blade towards the tip. The blade was excited three times for each measurement phase.

Results

For each of the two tests the modal parameters are determined, i.e. frequency, damping and mode shape. The figures show the comparison of the three first modes of the blade. In each graph is a series for the test with 0 deg. error on the accelerometers and for the test with 5 deg. error on the acc.

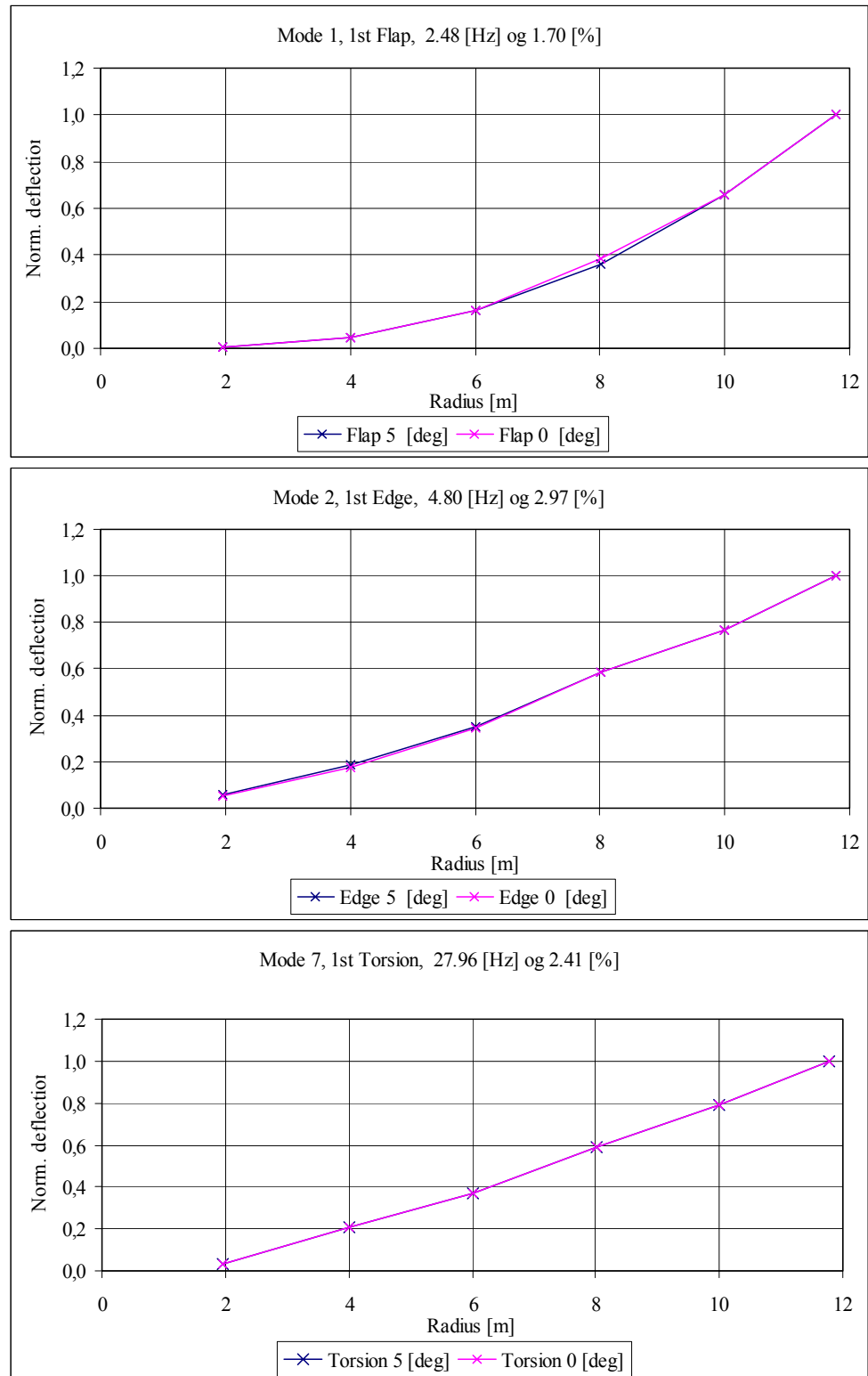


Figure 6. Comparison of results with (5°) and without (0°) angular error on accelerometers

Accelerometer mounting

Different equipment has been used for mounting the accelerometers on the blade. Figure 7 shows the types in consideration during this project. From left to right the types are a plastic device for mounting web-cameras, a ball/cone mounting clip from Brüel & Kjær, an aluminum camera ball head and a piece of angular steel. The bases of the adapters are attached to the blade by use of adhesive.

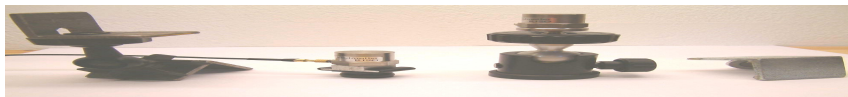


Figure 7. Different adapters for mounting accelerometers on the blade

The advantage of the first three adapters is the possibility of adjusting the direction of the accelerometer after the adapter has been mounted on the blade. With the angular steel plate directions have to be adjusted as the plate is mounted i.e. three planes and one position must be right.

The disadvantage of the first and third adapter is the high weight of the unit. This changes the properties of the blade. There is also a possibility of noise induced by resonance of the units.

The angular steel plate gives the possibility of mounting two accelerometers with two directions perpendicular to each other on the same adapter.

Conclusion

Two tests have been carried out with a difference in the measuring direction of the trailing edge accelerometers. In the second test these accelerometers were mounted with the active direction 5° off the horizontal plane.

The results of the two tests showed very little difference in the modal parameters. This indicates that mounting the accelerometers with the active direction within accuracy of 5° is sufficient for having useable results.

The tests and the properties of the different accelerometer adapters make the angular steel plate the most useable choice.

5 Test methods

5.1 Excitation techniques

The blade can be excited by two different techniques, either transient excitation or continuous excitation. A hydraulic or an electromagnetic exciter carries out continuous excitation by inducing a random, sinusoidal or white noise signal to the tested structure in the frequency range of interest.

The transient excitation is carried out by use of a hammer. The hammer induces an impulse force to the structure.

One disadvantage of the continuous excitation technique is that exciters might have difficulties exciting frequencies below 1.5 Hz. Large blades have natural frequencies below this value.

By that reason only transient excitation is investigated in these tests.

Description of test

To evaluate the excitation technique two types of transient excitation tests were carried out, one by use of hand-held hammer and one by use of a construction mounted with a pendulum hammer.

The pendulum hammer is a modified proto type from a previous modal-project [1].

Pendulum hammer

The pendulum hammer makes it possible to excite the structure with very similar impulses for every repetition of the excitation. The hammer is working in a defined plane as it turns around a fixed axis. This is assuring the same angle of attack for every repetition. The hammer is pulled back to a certain position, and is turning off that point for every impulse. This ensures the same magnitude of impulse for all impulses.

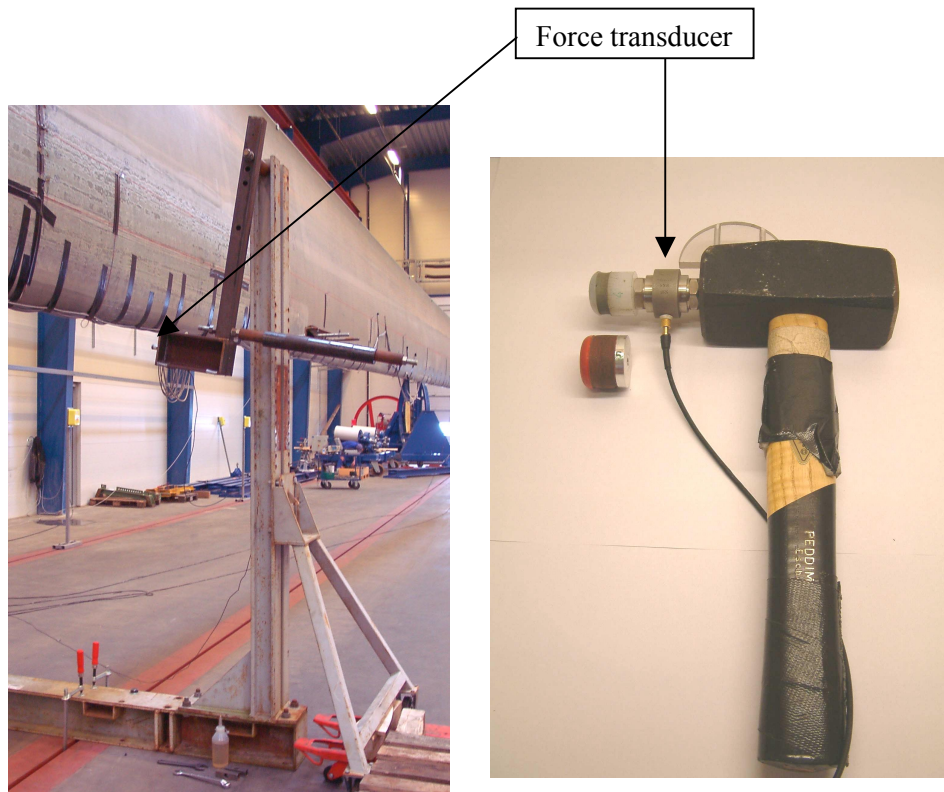


Figure 8. Pendulum hammer and hand-held hammer

Hand-held hammer

The advantage of the hand-held hammer is the portability and the possibilities of exciting structures in narrow areas.

Using the hand-held hammer gives some troubles in controlling that both the magnitude and the direction of the impulse is repeated for every excitation.

The test is expected to show that higher similarity in the induced impulses will give better modal parameter estimation.

Experimental test set-up

The blade used for test was the Nedwind 25 blade.

The tested blade was mounted in the test rig with the trailing edge upwards. The blade was discretized in six cross-sections, i.e. 18 DOF's.

Cross-Section		Z	X	Y	Active direction
	DOF	[m]	[m]	[m]	
1	1	1.97	-0.95	0.08	Y
1	3	1.97	0.35	-0.17	X
1	2	1.97	0.35	-0.17	Y
2	4	4.00	-0.90	0.03	Y
2	6	4.00	0.36	-0.15	X
2	5	4.00	0.36	-0.15	Y
3	7	6.00	-0.78	-0.03	Y
3	9	6.00	0.31	-0.16	X
3	8	6.00	0.31	-0.16	Y
4	10	8.01	-0.66	-0.09	Y
4	12	8.01	0.25	-0.17	X
4	11	8.01	0.25	-0.17	Y
5	13	9.99	-0.54	-0.15	Y
5	15	9.99	0.19	-0.19	X
5	14	9.99	0.19	-0.19	Y
6	16	11.79	-0.36	-0.19	Y
6	18	11.79	0.14	-0.19	X
6	17	11.79	0.14	-0.19	Y

The blade was excited in both edgewise and flapwise direction simultaneously by impact of the modal-hammer mounted with the force transducer. The hammer with force transducer was set up to excite the blade in a distance of 7 meters from the root intersection under an angle of 45° to both the flapwise and edgewise direction. The point of attack was on the suction side of the leading edge. The force direction was perpendicular to the Z-direction of the blade.

Modal test

The hand-held hammer test and the pendulum hammer test were carried out in three phases with measurements in two cross-sections in each phase. The measurements were made from the root end of the blade towards the tip. The blade was excited three times for each measurement-phase. When the impulse was induced the blade was oscillating in free vibrations until the vibrations died out. When the blade is not oscillating more a new impulse was induced to the blade.

Results

The following section discusses the results, and three first mode shapes (first flapwise, first edgewise and first torsion mode) are presented along with two second (flap and edge) modes.

The results show that the tests with the pendulum hammer seem to give more harmonic mode shapes. One of the reasons for this is higher accuracy between the single repetitions of excitation of the blade. The magnitude of the force and the direction is more exact from one excitation to the next.

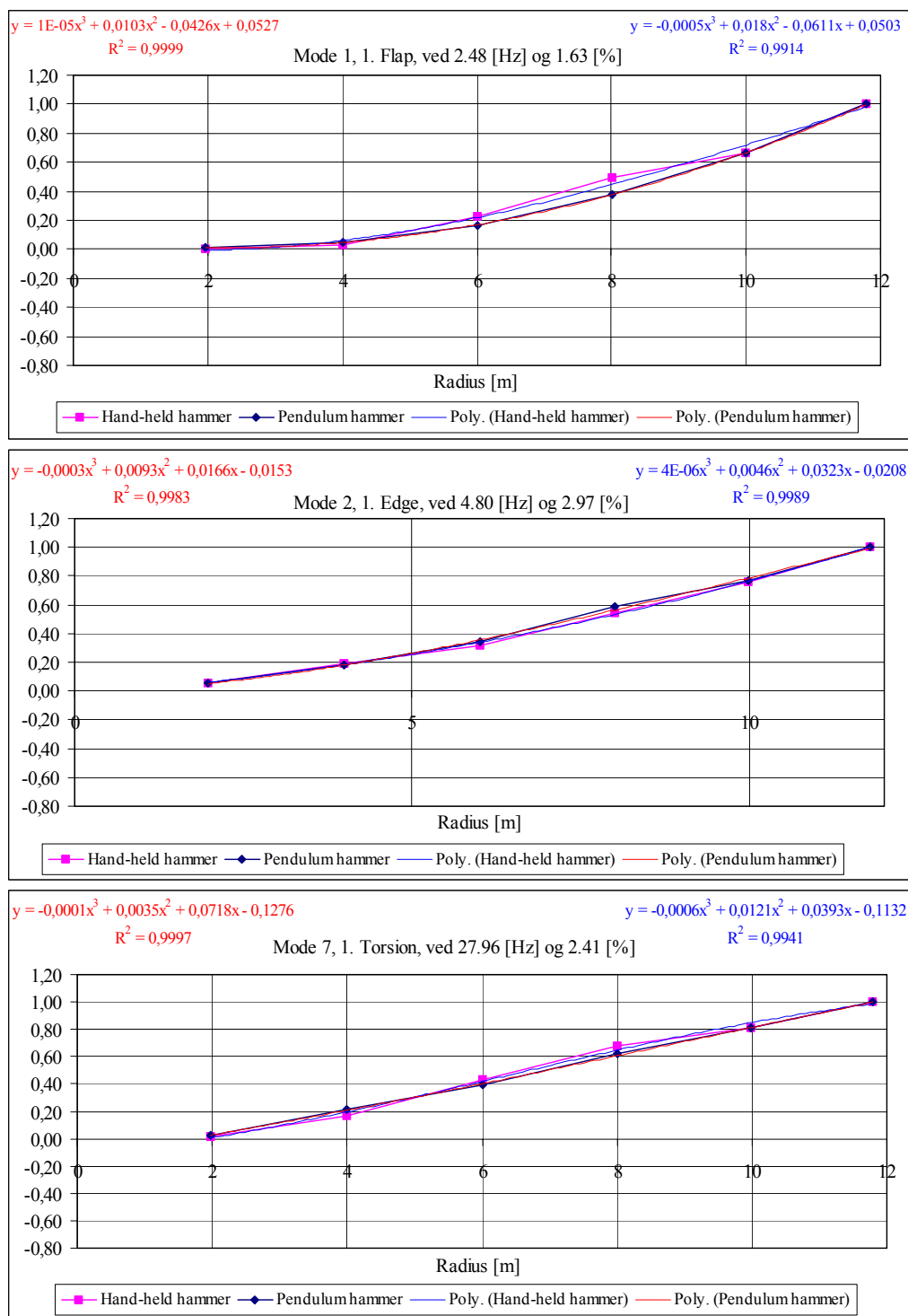


Figure 9. The three first-mode shapes, comparison of pendulum and hand-held hammer

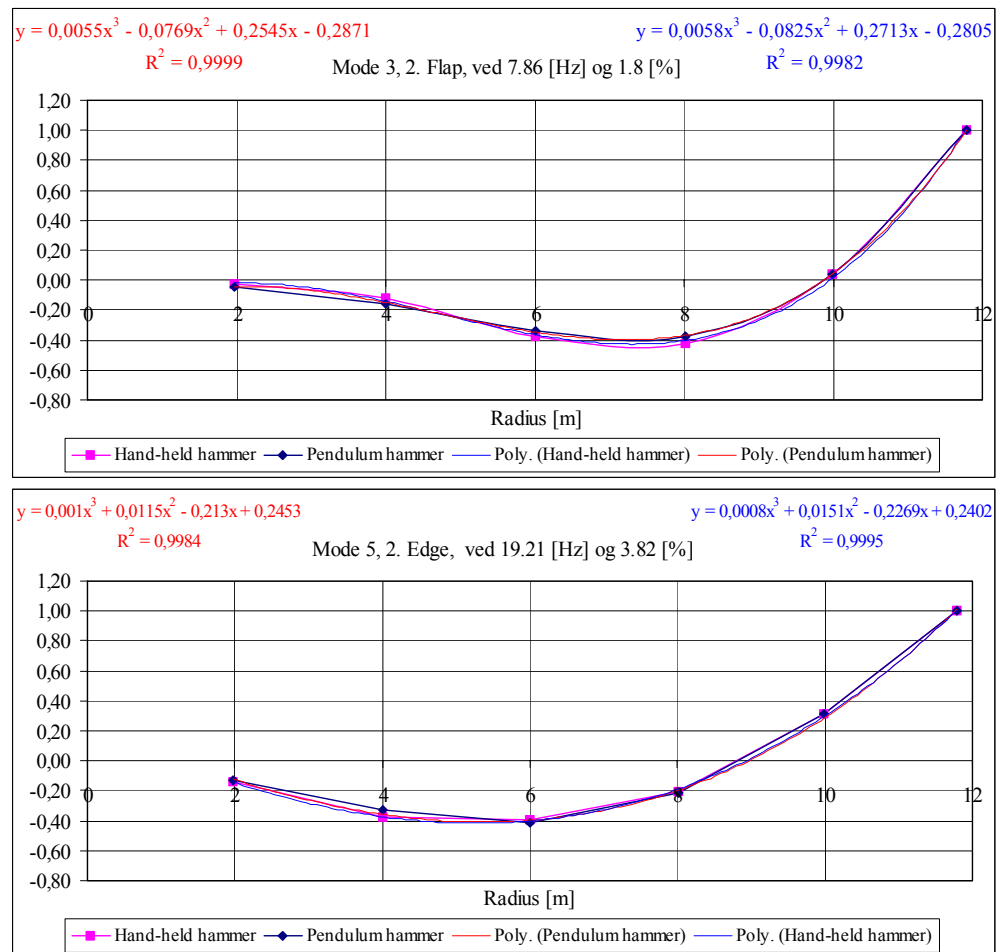


Figure 10. The two second-mode shapes, comparison of pendulum and hand-held hammer

Conclusion

The pendulum hammer gives higher accuracy in replication of the impact to the blade. The pendulum hammer has the same angle of attack for every impact and turns around the same axis for every impact. It is also seen that the data from the pendulum hammer test fits the chosen curve better than the hand-held hammer. Further tests have to be made to have a statistical basis to evaluate the two impact methods.

Excitation direction, flapwise or edgewise

The blade is excited in one direction, and the direction is chosen to excite the modes of interest. If only the modes in either flapwise or edgewise direction are of interest a better signal-to-noise ratio could be obtained with excitation in the particular direction.

Description of test

To evaluate the excitation directions two tests were carried out to compare the results of either vertical or horizontal excitation.

Experimental test set-up

The blade used for test was a Nedwind 25 blade.

The tested blade was mounted with the trailing edge upwards. The blade was discretized in six cross-sections, i.e. 18 DOFs.

Cross-Section		Z	X	Y	Active direction
	DOF	[m]	[m]	[m]	
1	1	1.97	-0.95	0.08	Y
1	3	1.97	0.35	-0.17	X
1	2	1.97	0.35	-0.17	Y
2	4	4.00	-0.90	0.03	Y
2	6	4.00	0.36	-0.15	X
2	5	4.00	0.36	-0.15	Y
3	7	6.00	-0.78	-0.03	Y
3	9	6.00	0.31	-0.16	X
3	8	6.00	0.31	-0.16	Y
4	10	8.01	-0.66	-0.09	Y
4	12	8.01	0.25	-0.17	X
4	11	8.01	0.25	-0.17	Y
5	13	9.99	-0.54	-0.15	Y
5	15	9.99	0.19	-0.19	X
5	14	9.99	0.19	-0.19	Y
6	16	11.79	-0.36	-0.19	Y
6	18	11.79	0.14	-0.19	X
6	17	11.79	0.14	-0.19	Y

The blade was excited in two experimental campaigns, one in edgewise and one in flapwise direction. The hammer mounted with a force transducer was set up to excite the blade in a distance of 7 meters from the root intersection. In flapwise direction the point of attack was on the beam on the suction side of the blade. In the edgewise direction the blade was excited on the leading edge (see Figure 11). For both tests the force direction was perpendicular to the Z-direction of the blade.



Figure 11. Vertical excitation of the blade

Modal test

Both the edgewise and the flapwise test were carried out in three phases with measurements in two cross-sections in each phase. The measurements were made from the root end of the blade towards the tip. The blade was excited three times for each measurement-phase. When the impulse was induced the blade was oscillating in free vibrations until the vibrations died out. When the blade is not oscillating more a new impulse was induced to the blade.

Results

The following section discusses the results, and the two first mode shapes (1. flapwise, 1. edgewise mode) are presented in comparison with the results from a combined test (45° angle of load introduction).

Figure 12 and Figure 13 show no difference in the mode shapes of the blade whether the excitation direction is horizontal, vertical or combined. The differences between the three results are within the statistical variation in replicated tests.

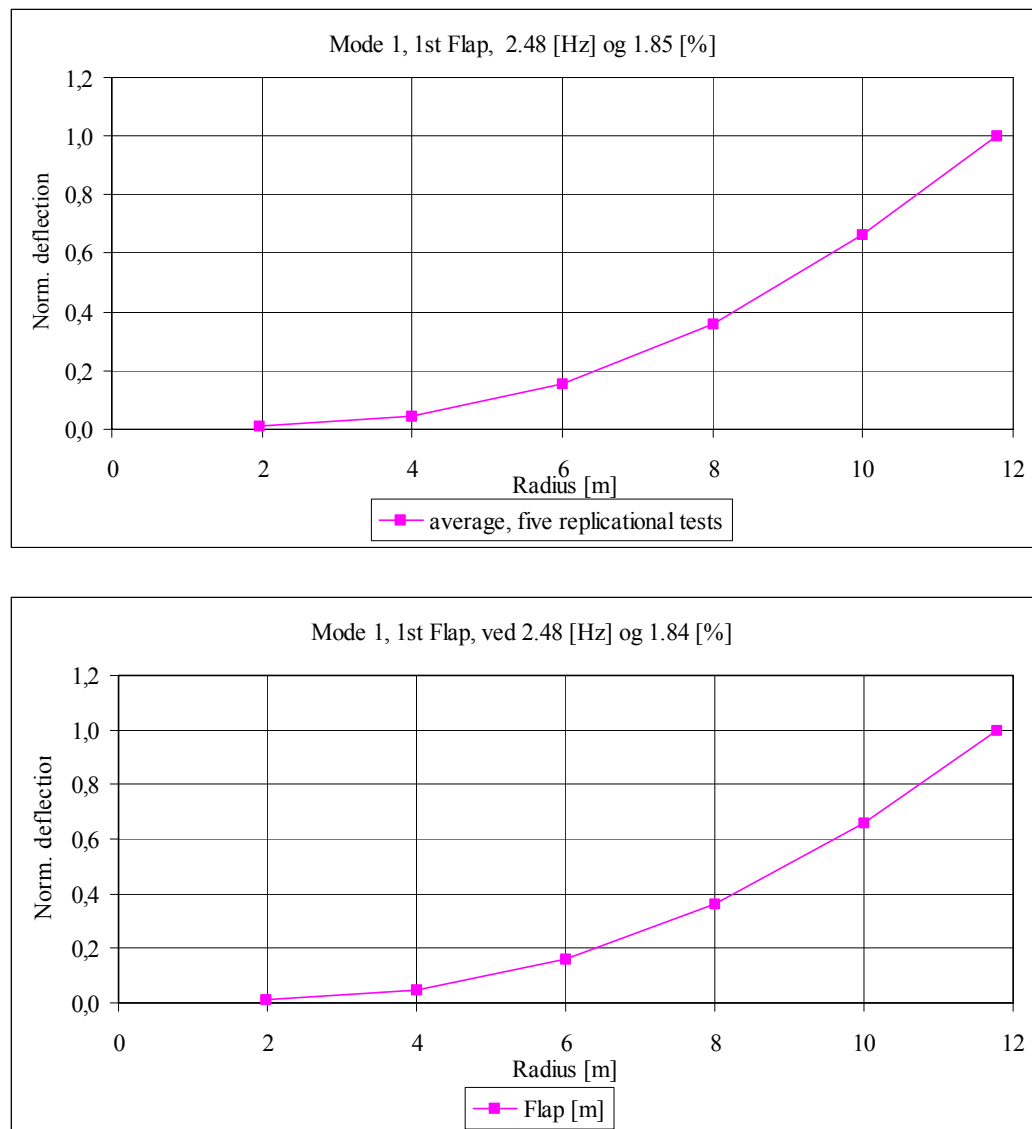


Figure 12. Comparison of first flapwise mode shape. The upper graph shows the mode shape when the blade is excited in both flapwise and edgewise direction simultaneously. In the lower graph the blade is excited in only flapwise direction

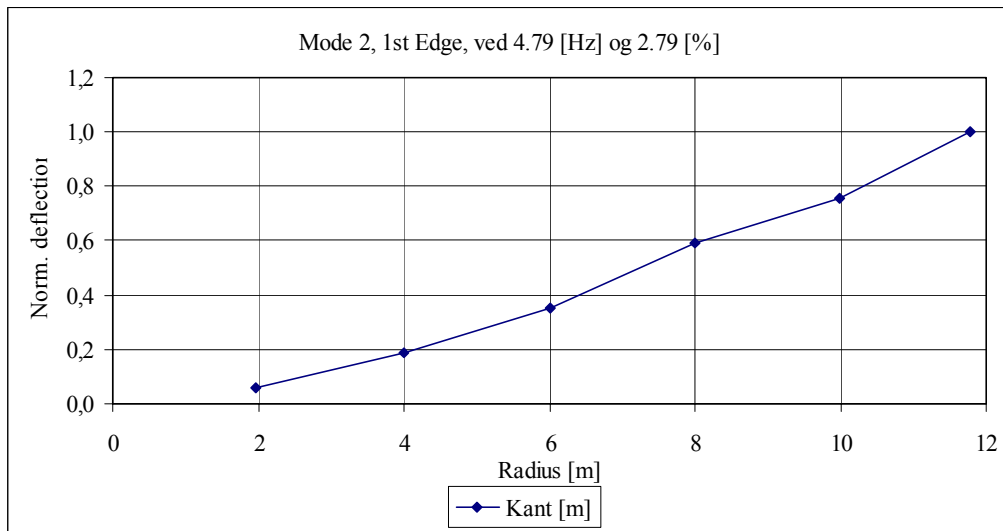
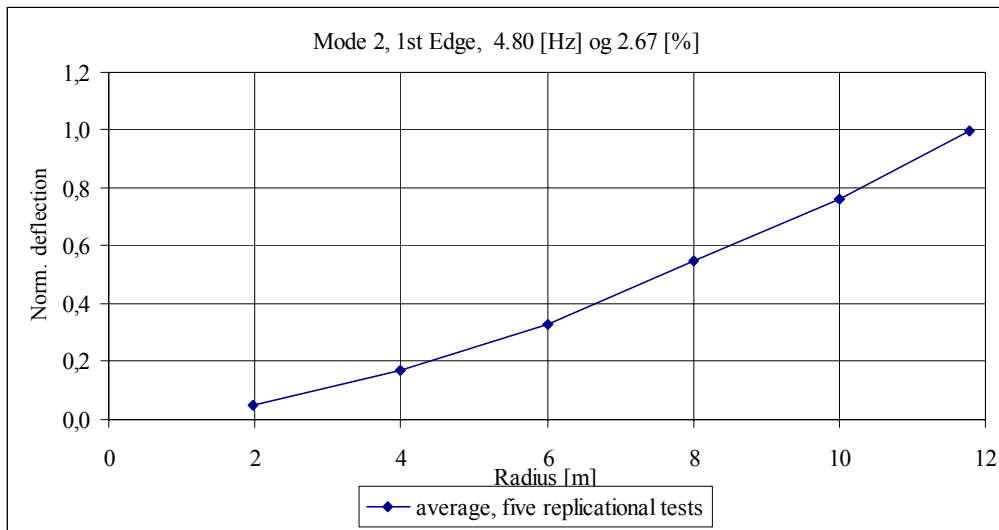


Figure 13. Comparison of first edgewise mode shape. The upper graph shows the mode shape when the blade is excited in both flapwise and edgewise direction simultaneously. In the lower graph the blade is excited in only edgewise direction

Conclusion

When modal analysis is carried out on structures as turbine blades it seems as a combined direction of load introduction, an angle of 45° to the suction/pressure surface of the blade, gives acceptable results. If a certain mode is difficult to excite by this approach, excitation in the direction of this particular mode can have improving effect on the results.

New excitation hammer

Due to the higher accuracy in the replication of the test by use of the pendulum hammer it was decided to improve/re-design the used pendulum hammer. This re-design of the hammer led to a new concept. Instead of the pendulum hammer two linear electro motors in series applies the excitation force. To keep the possibility to excite the test specimen in either vertical or horizontal direction the new hammer is equipped with a guidance that makes the excitation direction adjustable. The new exciter is carried out in a combination of steel and aluminum to have a strong and light unit.



Figure 14. Re-designed impact hammer with linear electro motor excitation.

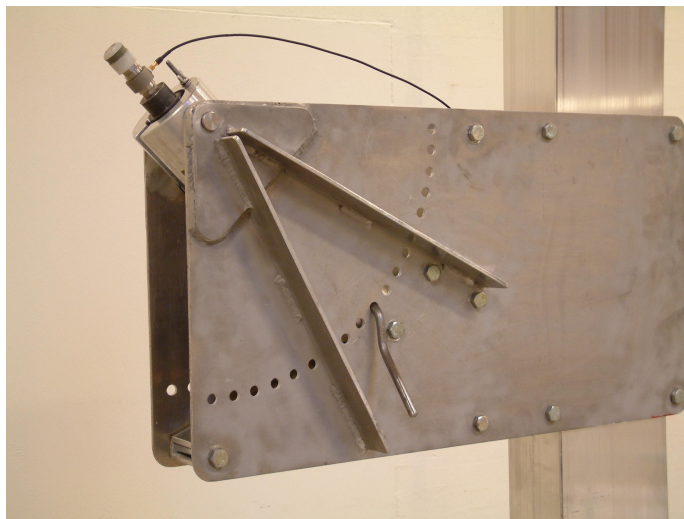


Figure 15. Close-up of exciter. This set-up gives the possibility to excite the blade in any direction from vertical to horizontal.

5.2 Replication of test to verify accuracy

Purpose

The present chapter deals with the results obtained when the modal analysis is repeated five times on a wind turbine blade. The present experimental investigation aims at comparing the results from five modal analyses conducted on the same wind turbine blade. Furthermore the frequency and damping results will be compared with traditional frequency analysis.

Experimental procedures

The experimental tests have been performed on a 12 m Nedwind NW25 wind turbine blade mounted with the tip chord in a vertical position by clamping the root blade flange to a rigid test rig. To prepare for a hammer exciting forces (forces with an angle of the magnitude of 45 deg with the chord that is able to excite flapwise as well as edgewise dominated blade modes simultaneously) the blade is for practical reasons mounted with the robust leading edge pointing downwards.

The transient force is established by means of a pendulum hammer that ensures sufficient accuracy in the location and direction, furthermore the device have sufficient mass and velocity to impacts the wind turbine blade once giving it an initial velocity and acceleration. The structure is at rest before the excitation. After the hammer impact the external force is zero again.

Note that the location of the force must not coincidence with nodes of modes of interest and that the direction of the applied forcing should be adjusted to the mode shapes of relevance for the investigation. Usually one experimental campaign is enough to resolve all modes shapes of interest.

The hammer loading is recorded by means of a B&K force transducer mounted on a hammerhead, and with a mechanical rubber head mounted on it. The characteristics of the mechanical rubber head must be adjusted based on a trial and error procedure, until the wanted input frequency characteristics are achieved.

Experimental test

When applying the forcing care should be taken to achieve impulse loadings of approximately the same magnitude for repeated strokes related to a particular cross-section in order to improve the signal / noise ratio. For the same reason care should also be taken to obtain loadings of a suitable magnitude. Only three repeated strokes are performed in the present experimental campaign.



Figure 16. Experimental set-up showing the blade mounted on the test rig

Results

The natural frequencies obtained from five modal analysis measurements are presented in Table 4. Natural frequencies are determined from three acceleration recordings in each cross section. The values for the natural frequencies given in the table are the average values computed based on all available recordings.

Table 4. Natural frequencies obtained from five modal analysis measurements

Number of Measurement	Mode						
	frequency						
	1. flap	1. edge	2. flap	3. flap	2. edge	4. flap	1. torsion
1	2.480	4.799	7.851	15.968	19.246	25.212	27.988
2	2.480	4.792	7.849	15.966	19.252	25.204	27.982
3	2.479	4.795	7.848	15.964	19.241	25.200	27.980
4	2.478	4.795	7.847	15.961	19.251	25.201	27.975
5	2.479	4.796	7.847	15.961	19.245	25.198	27.974
Average	2.479	4.795	7.848	15.964	19.247	25.203	27.980
Std. dev.	0.001	0.002	0.001	0.003	0.004	0.005	0.005

The natural frequencies obtained from a traditional measurement are presented in Table 5. Natural frequencies are obtained by one accelerometer mounted onto the centerline of the blade at the tip. The blade was excited into its natural frequency by hand and then allowed to oscillate free. The accelerometer signal was amplified and recorded in the computer. The torsional frequency was determined using the differential signal from two accelerometers placed on the leading and trailing edge respectively. The blade was excited by hand into its torsional mode and then allowed to oscillate free. The signal was lead to a digital sample oscilloscope, stored, and recorded into a file.

Table 5. Natural frequencies obtained from one traditional measurement

Number of measurement	Mode						
	frequency						
	1. flap	1. edge	2. flap	3. flap	2. edge	4. flap	1. torsion
1	2.52	4.89	8.00	NA	NA	NA	28.60

The damping characteristics are derived from the modal analysis as expressed by the logarithmic decrement. Determination of the damping properties is usually considered to be uncertain due to the fact that the damping characteristics

are small quantities. The damping obtained from modal analysis measurements are presented in Table 6.

Table 6. Damping obtained from five modal analysis measurements

Number of measurement	Mode						
	Damping (log. dec. %)						
	1. flap	1. edge	2. flap	3. flap	2. edge	4. flap	1. tor-sion
1	1.650	2.350	1.740	1.910	3.590	2.55	2.340
2	2.050	2.520	1.780	1.960	3.650	2.53	2.350
3	2.200	2.370	1.820	2.000	3.610	2.55	2.410
4	1.680	2.510	1.790	1.990	4.200	2.57	2.370
5	1.650	2.590	1.770	1.990	3.730	2.55	2.370
Average	1.846	2.468	1.780	1.970	3.756	2.55	2.368
Std. dev.	0.233	0.093	0.026	0.033	0.227	0.013	0.024

The damping obtained from traditional measurements are presented in Table 7.

Table 7. Damping obtained from traditional measurements

Number of measurement	Mode						
	Damping (log. dec. %)						
	1. flap	1. edge	2. flap	3. flap	2. edge	4. flap	1. tor-sion
1	2.3	2.4	NA	NA	NA	NA	NA

The mode shape results associated with the natural frequencies are illustrated in Figure 17 to Figure 23. For each mode only the dominating modal deflection is presented. To illustrate the deviation in the measured mode shapes in the measuring campaign, each graph present the maximum, minimum and the average value of five measurements.

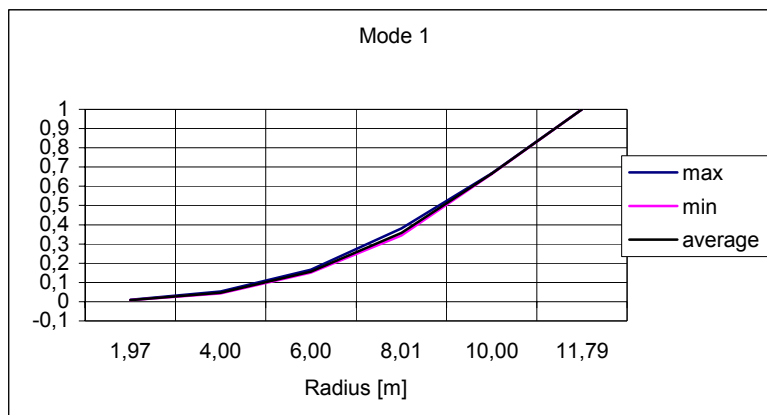


Figure 17. Mode shape for mode 1, presented as maximum, minimum and average

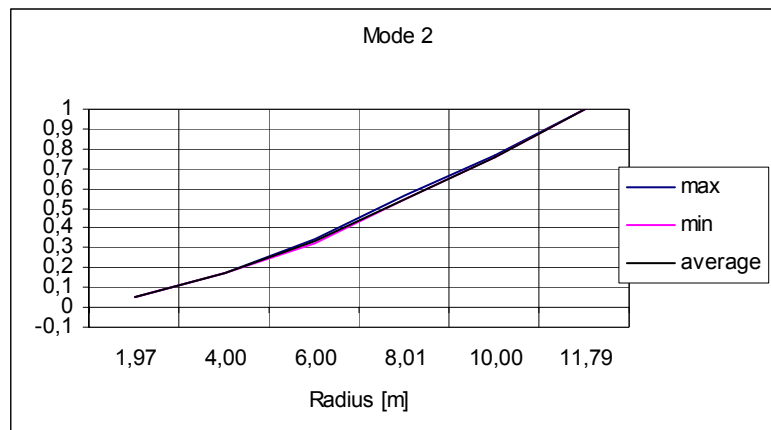


Figure 18. Mode shape for mode 2, presented as maximum, minimum and average

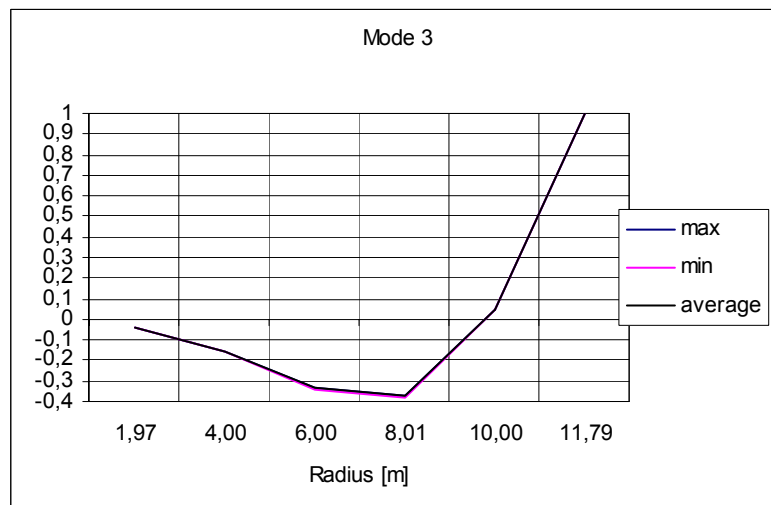


Figure 19. Mode shape for mode 3, presented as maximum, minimum and average

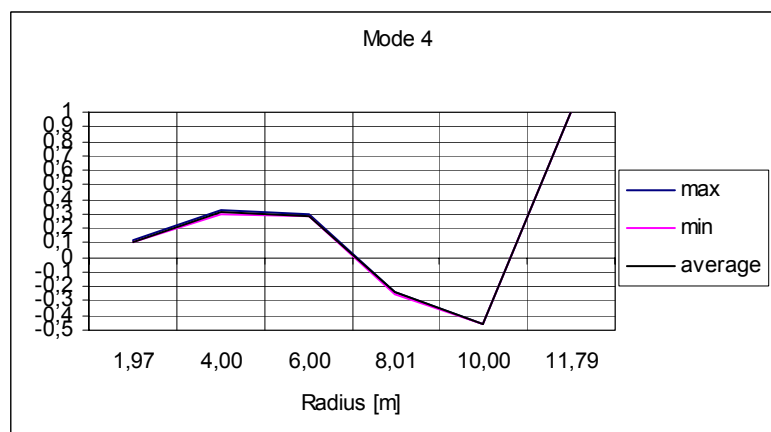


Figure 20. Mode shape for mode 4, presented as maximum, minimum and average

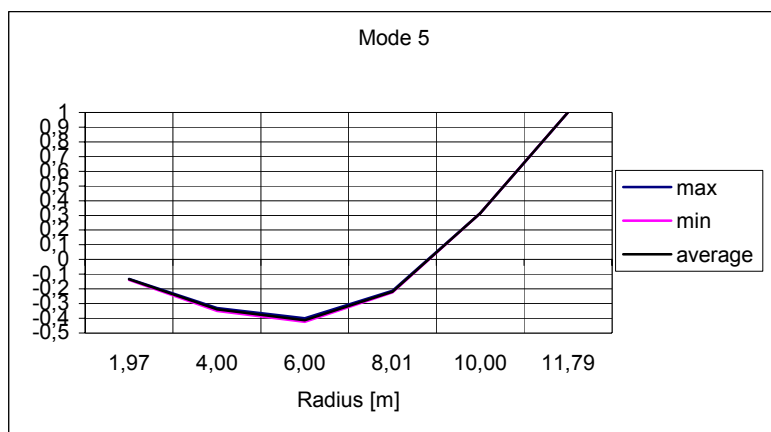


Figure 21. Mode shape for mode 5, presented as maximum, minimum and average

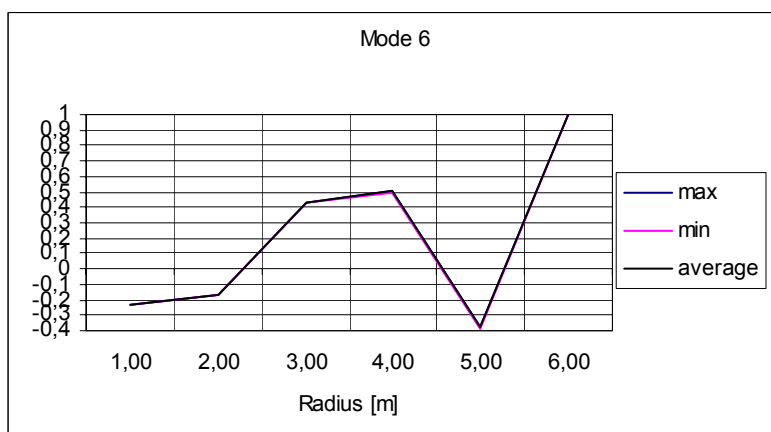


Figure 22. Mode shape for mode 6, presented as maximum, minimum and average

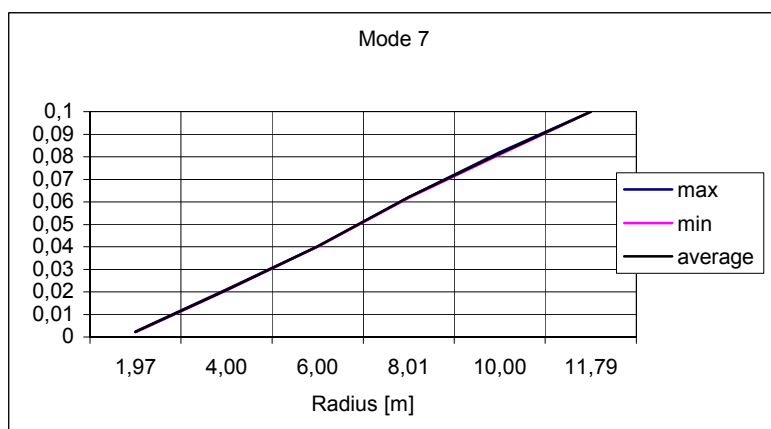


Figure 23. Mode shape for mode 7, presented as maximum, minimum and average

Conclusion

This chapter presents results from five modal analyses conducted on the same wind turbine blade. The aim was to determine the deviation between the five analyses. Concerning the frequencies there were a good agreement when comparing the results. Apparently is the deviation in the results larger at higher frequencies. Concerning the damping there are deviation in the results independent of the frequency. This is due to the fact that the damping characteristics are small quantities. As seen in Figure 17 to Figure 23 the mode shapes are very similar when the measurements are repeated. The obtained results are reliable and the procedure with transient excitation by use of the pendulum hammer and measuring out-put in two cross-section simultaneously seems to be reliable as well.

6 Stiffness of the set-up

Description of test

The results from modal analysis performed on a blade are depending on the fixture of the blade. The stiffness of the test-rig and the stiffness of the adapter between blade and test-rig are items to be precautionary with. A weak fixture may cause lower frequencies and damping. The mode shapes of the tested blade are also depending on the fixture. The position of the nodes in the mode shapes may change with change in the stiffness of the fixture.

To determine the influence of the stiffness of the fixture a test has been carried out.

The test is carried out in combination with a traditional test, i.e. with the blade mounted in a test rig with the tip-chord in vertical direction and the leading edge pointing down-wards. A standard modal analysis is done by discretize the blade and measure accelerations in three DOF's for each section. The first section is normally positioned in some distance off the root-interface.

Modal test of blades show that very little or no movement is observed in the three measured DOF's in the section closest to the root-interface.

The purpose of the test is to determine which DOF's that are influenced by the deflection of the fixture.

Thus the movement of the root intersection is considered negligible in the flap-wise and edgewise direction as well as the torsional direction, only the longitudinal movement of the blade in the root interface was measured. It is expected that a first or second mode of the mounting plate will be excited by excitation of the blade.

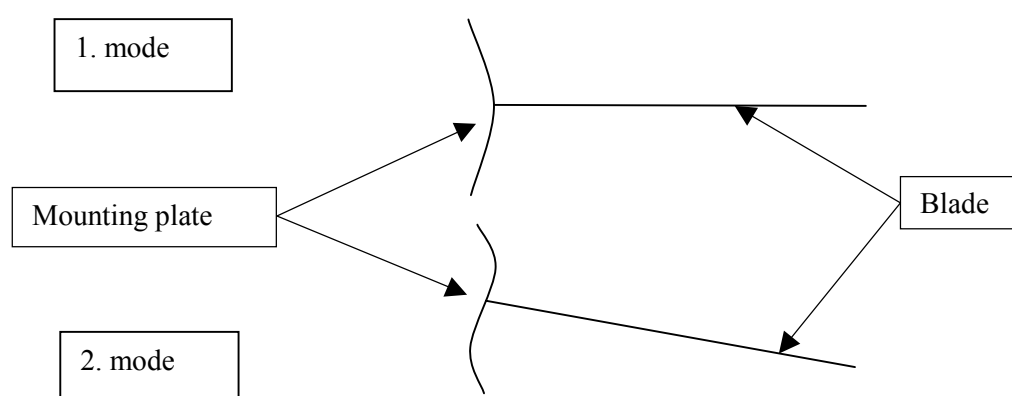


Figure 24. Sketch of test set-up, showing the expected modes of the mounting plate

The mounting plate is a circular plate and both first and second mode of the plate are expected to be possible to excite in the flapwise and the edgewise direction of the blade.

Test set-up

The tested blade was mounted in a test rig with the trailing edge up-wards and the tip-chord in vertical position. The blade was discretized in eight sections for the determination of modal parameters of the blade. To determine the movement of the fixture additionally two sections were defined. These two sections were positioned in 16 % of total blade length and in the root interface.

The graph in Figure 25 shows the discretization of the blade.

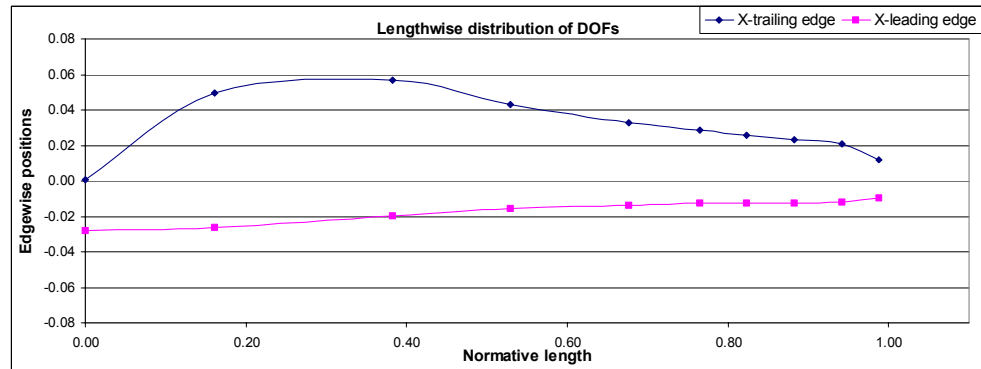


Figure 25. Lengthwise distribution of accelerometer positions

The section in the root-interface is instrumented with one accelerometer on the leading edge and one accelerometer on the pressure side and suction side respectively. These three accelerometers are all measuring in the Z-direction, i.e. the longitudinal direction of the blade. In the other sections one accelerometer is measuring in the direction of the chord (edgewise) and two accelerometers are measuring in the flapwise direction. The torsional accelerations are expressed as the difference between the two flapwise accelerometers.

The fixture is made of a circular steel plate with a diameter of approximately 2.8 meters and with a BCD of the root interface on the blade of app. 1.8 meters. The plate has a thickness of 90 millimeters.

The excitation of the structure is carried out by transient excitation by use of a hammer on which a force transducer is mounted. The force transducer and hammer is positioned to have an angle of attack of 45° off horizontal and perpendicular to the longitudinal axis of the blade. Thus the blade is positioned with the tip chord vertical, both edgewise and flapwise modes are excited by this direction of the impulse. The measurements are carried out with six accelerometers, which means that two sections are tested simultaneously. For every two sections the blade is excited three times. The excitation is an impulse applied to the blade in app. 80% of blade length. When the impulse is applied to the blade the blade is oscillating in free vibration, i.e. it vibrates in the mode shapes of the blade. When the oscillations have run out and the blade is at rest, a new impulse is applied to the blade.

Results

To describe the results the magnitude of the measured signals are normalized for each mode shape. The highest value in each mode is set to 1. In a plot of the normative accelerometer signals as function of the length of the blade it is seen that in the mode shapes of interest, only signals of negligible magnitude are measured in the root interface, see Figure 26. **NOTE:** The graphs (Figure 26

and Figure 27) shows the magnitude of the accelerometer signals only, there are no directions on the vectors!

The signals in the root interface are of magnitude 0.5 – 1.0 per thousand of the highest measured value.

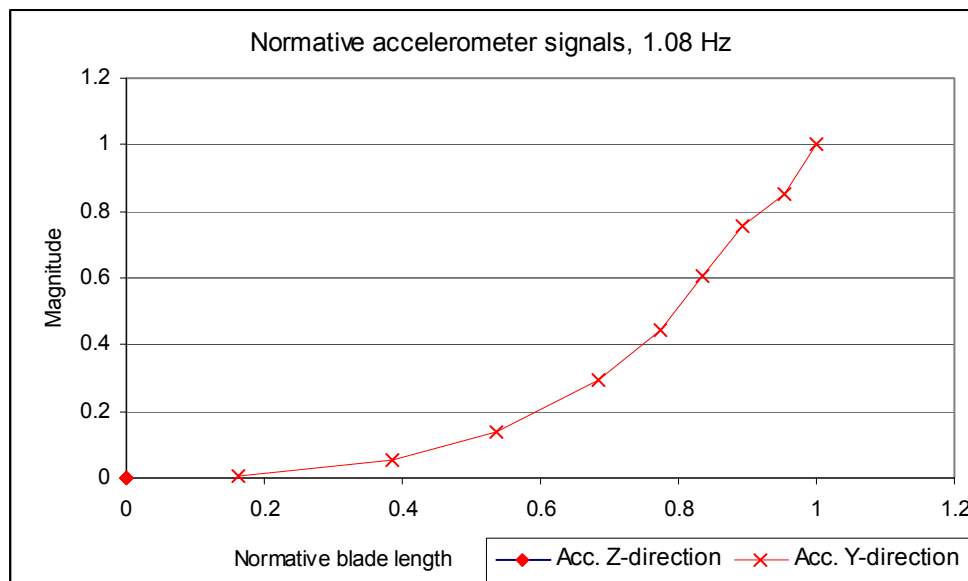


Figure 26. Normative accelerometer magnitude as function of blade length, first flap-wise mode

For this particular test set-up, the picture from the first flap-mode (Figure 26) rules for the natural frequencies normally measured on the blades, i.e. first to fourth flapwise, first and second edgewise and first torsional natural frequency.

In some mode shapes it is possible to excite the mounting plate. For a natural frequency at 20.01 Hz the accelerometer signals at the adapter plate are of magnitude 45 per cent of the maximum signal. The resonance is a combination of a flapwise and edgewise mode of the blade and a movement of the mounting plate in the longitudinal direction of the blade, cf. Figure 24, 1. mode of mounting plate.

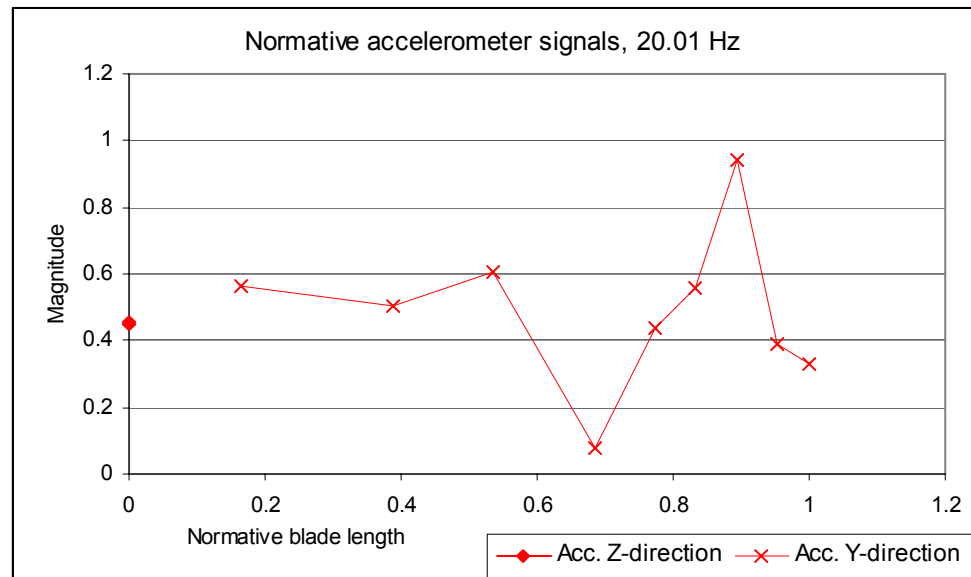


Figure 27. Normative accelerometer magnitude as function of blade length.

Conclusion

Whether the fixture will influence on the mode shapes of interest or not is depending on the natural frequency of the test-rig and the fastening of the blade to the rig. As the measured frequencies are very low and the system of fixture has a higher natural frequency it is believed that the fixture is without importance for the measured values.

In the present test only the movement of the adapter plate has been measured, whether the test rig has moved or not is not known. It is assumed that the test rig is rigid and the measured deflection is regarding only the mounting plate.

If the natural frequency of the fixture is unknown and a thin (flexible) adapter plate is used to mount a heavy blade, precautions must be taken to ensure that the natural frequency of the blade does not interfere with the natural frequency of the system of fixture.

7 Modal analysis on loaded and unloaded blade

Purpose

The present chapter deals with the results obtained when the modal analysis is conducted on the wind turbine blade situated in one of two positions. This is in order to determine the difference in the modal analysis results when the blade is influenced by its net weight due to the gravity and not. During the test program the wind turbine blade is situated in two positions. First it is mounted on the test rig in a horizontal position with the tip chord in a vertical position. Afterwards the wind turbine blade is mounted in a vertical position with the tip chord in a horizontal position. The present experimental investigation aims at comparing the results from these two experiments, i.e. frequency, damping and the mode shapes.

Experimental procedures

The experimental tests have been performed on a 7.5 m wind turbine blade. To prepare for a hammer exciting forces (forces with an angle of the magnitude of 45 deg with the chord that is able to excite flapwise as well as edgewise dominated blade modes simultaneously) the blade is for practical reasons mounted with the robust leading edge pointing downwards, when the blade is mounted in a horizontal position. When the blade is mounted horizontally the blade is excited in a distance of 4.5 m from the blade root interface.

Then the blade is mounted in a vertical position the hammer exciting force still is of a magnitude of 45 deg with the chord that is able to excite flapwise as well as edgewise dominated blade modes simultaneously. When the blade is mounted vertically the blade is excited in a distance of 2.0 m from the blade root interface.

The transient force is established by means of a hammer that ensures sufficient accuracy in the location and direction, furthermore the device have sufficient mass and velocity to impacts the wind turbine blade once giving it an initial velocity and acceleration. The structure is at rest before the excitation. After the hammer impact the external force is zero again.

Note that the location of the force must not coincident with nodes of modes of interest and that the direction of the applied forcing should be adjusted to the mode shapes of relevance for the investigation. Usually one experimental campaign is enough to resolve all modes shapes of interest.

The hammer loading is recorded by means of a B&K force transducer mounted on a hammerhead, and with a mechanical rubber head mounted on it. The characteristics of the mechanical rubber head must be adjusted based on a trial and error procedure, until the wanted input frequency characteristics are achieved.

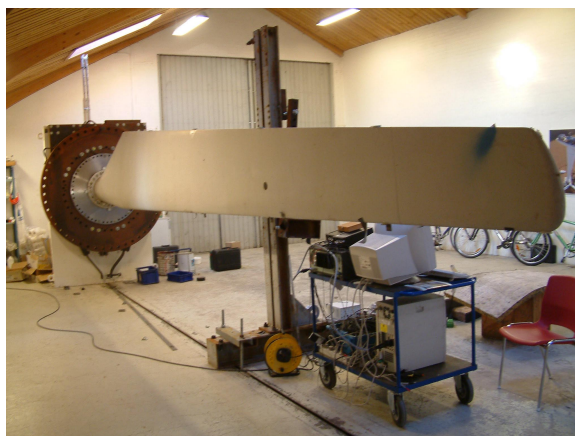


Figure 28. Experimental set-up showing the blade in horizontal position. The hammer is behind the blade.



Figure 29. Experimental set-up showing the blade in vertical position. The hammer is behind the blade.



Figure 30. Hammer set-up, when the blade is in vertical position.

Experimental test

When applying the forcing care could be taken to achieve impulse loadings of approximately the same magnitude for repeated strokes related to a particular cross-section in order to improve the signal / noise ratio. For the same reason care should also be taken to obtain loadings of a suitable magnitude. Only three repeated strokes are performed in the present experimental campaign. After each stroke the blade is oscillating in free vibration until the vibration runs out. When the vibration has stopped, the next stroke is applied to the blade.

Results

The natural frequencies obtained for the wind turbine blade in a horizontal- and vertical position are presented in Table 8. Natural frequencies are determined from three acceleration recordings in each cross section. The values for the natural frequencies given in the table are the average values computed based on all available recordings.

Table 8. Natural frequency measurements presented for horizontal and vertical position of the blade

Number of Measurement	Blade direction	Mode				
		1. flap	1. edge	2. flap	3. flap	1. torsion
		Frequency				
1	Horizontal	2.83	6.24	10.36	20.02	27.88
2	Vertical	2.80	6.14	10.52	20.34	27.77

The damping for the wind turbine blade in horizontal- and vertical position respectively are presented in Table 9.

Table 9. Damping measurements presented for horizontal and vertical position of the blade

Number of Measurement	Blade direction	Mode				
		1. flap	1. edge	2. flap	3. flap	1. torsion
		Damping (log. dec. %)				
1	Horizontal	3.32	12.29	3.61	3.30	6.56
2	Vertical	3.28	9.27	3.19	5.15	7.66

The mode shape results associated with the present natural frequencies are illustrated in Figure 31 to Figure 35. For each mode only the dominating modal deflection is presented. To illustrate the deviation in the determined mode shapes each graph present the results for horizontal and vertical mounting of the blade.

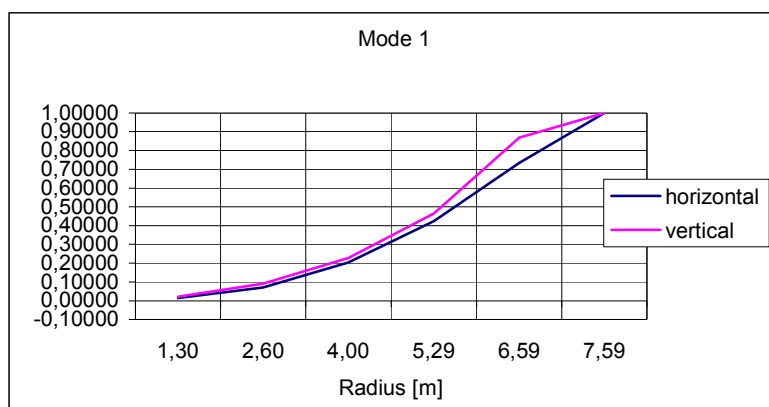


Figure 31. Mode shape for mode 1, 1. flapwise natural frequency.

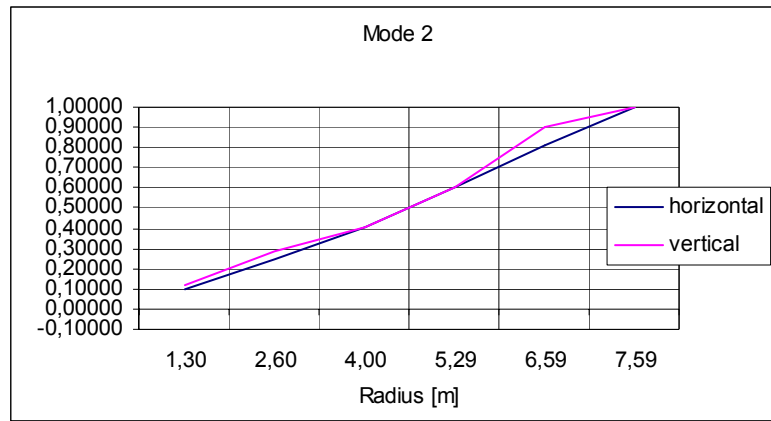


Figure 32. Mode shape for mode 2, 1. edgewise natural frequency.

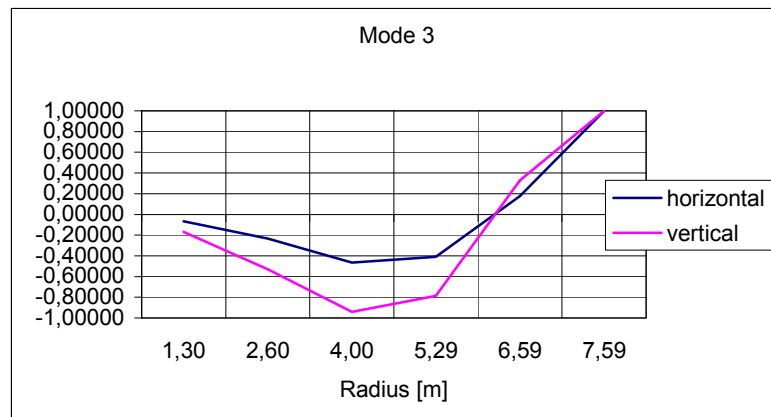


Figure 33. Mode shape for mode 3, 2. flapwise natural frequency.



Figure 34. Mode shape for mode 4, 3. flapwise natural frequency.



Figure 35. Mode shape for mode 5, 1. torsion natural frequency.

Blade mounting on the test rig

In the following the flexibility of the set-up is investigated. During the test campaign the flexibility at the root end of the wind turbine blade was measured by four accelerometers. They were mounted with an angle of 90 degree between each other and with the pitch axis at origin. They were numbered counter clock wise from one to four. Starting with number one in the + y direction. The accelerometers are measuring in the $-z$ direction. See Figure 36

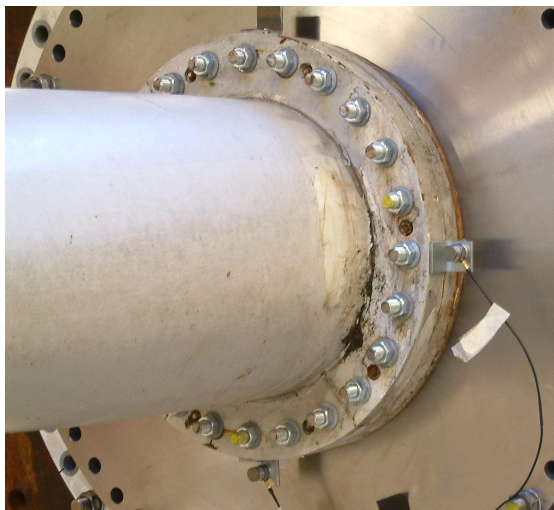


Figure 36. Experimental set-up for determination of the flexibility of the blade mounting on the test rig.

Figure 37 illustrates the deflection at the root end of the blade at the first edge-wise natural frequency of the blade. The figure shows that the test rig used for the vertical experiment is more flexible than the test rig used for the horizontal experiment at this frequency. Furthermore the figure shows that it is an edge-wise mode, because the largest displacement are seen at accelerometer number two and four, placed at trailing and leading edge respectively.

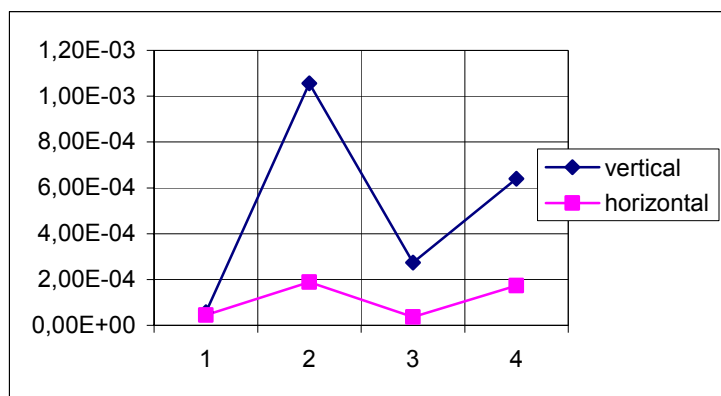


Figure 37. Deflection at the root end at 1. edgewise natural frequency 6.24 Hz

Figure 38 below illustrates the deflection at the root end of the blade at the second flapwise natural frequency of the blade. The figure shows that the test rig used for the horizontal experiment is more flexible than the test rig used for the vertical experiment. Furthermore the figure shows that it is a flapwise mode, because the largest displacement are seen at accelerometer number one and three, placed at suction and pressure side of the blade.

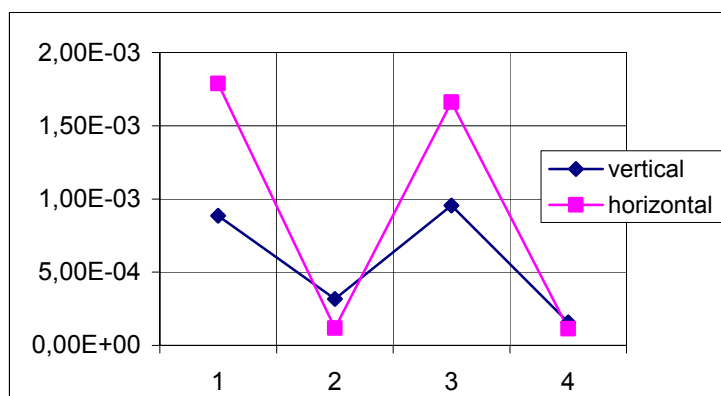


Figure 38. Deflection at the root end at 2. flapwise natural frequency 10.36 Hz

At the other natural frequencies of the blade there were not seen any significant movement of the fixture.

Conclusion

The orientation of the blade has an effect on the measured results obtained by modal analysis. There is a variation in the measured frequency on up till 1.6 %, at 1.edgewise and 3. flapwise natural frequencies. The damping results have a variation on up till 56%, at 3. flapwise natural frequency. The uncertainty on the damping is due to the fact that the damping characteristics are small quantities.

The investigation of the flexibility of the test rig shows that the magnitude of the deflection is dependent of the frequency. At 1. edgewise frequency the test rig used for the vertical experiment is more flexible than the test rig used for the horizontal experiment. At the 2. flapwise frequency the test rig used for the horizontal experiment is more flexible than the test rig used for the vertical experiment.

8 Experimental tests against computed models

Description of test

To compare the experimental results from the modal analysis with the results from a computer based model a test of a LM blade has been conducted. Prior to the experimental test LM carried out calculations of a model by use of Stig Øyes Blade_EV1 program. These calculations were used to decide where the accelerometers were positioned during the experimental test. The charts in Figure 39 show the first six mode shapes of the blade. In the upper chart of Figure 39 the flapwise mode shapes are normalized to 1, i.e. four of the mode shapes are mainly flapwise. In the lower chart the edgewise graphs are normalized to 1. The graphs which are not normalized to 1 in the flapwise chart corresponds to the graphs normalized to 1 in the edgewise chart. I.e. the main deflection direction is normalized to 1 and for the same shape the corresponding deflection in the opposite direction, or the perpendicular direction, is scaled relatively.

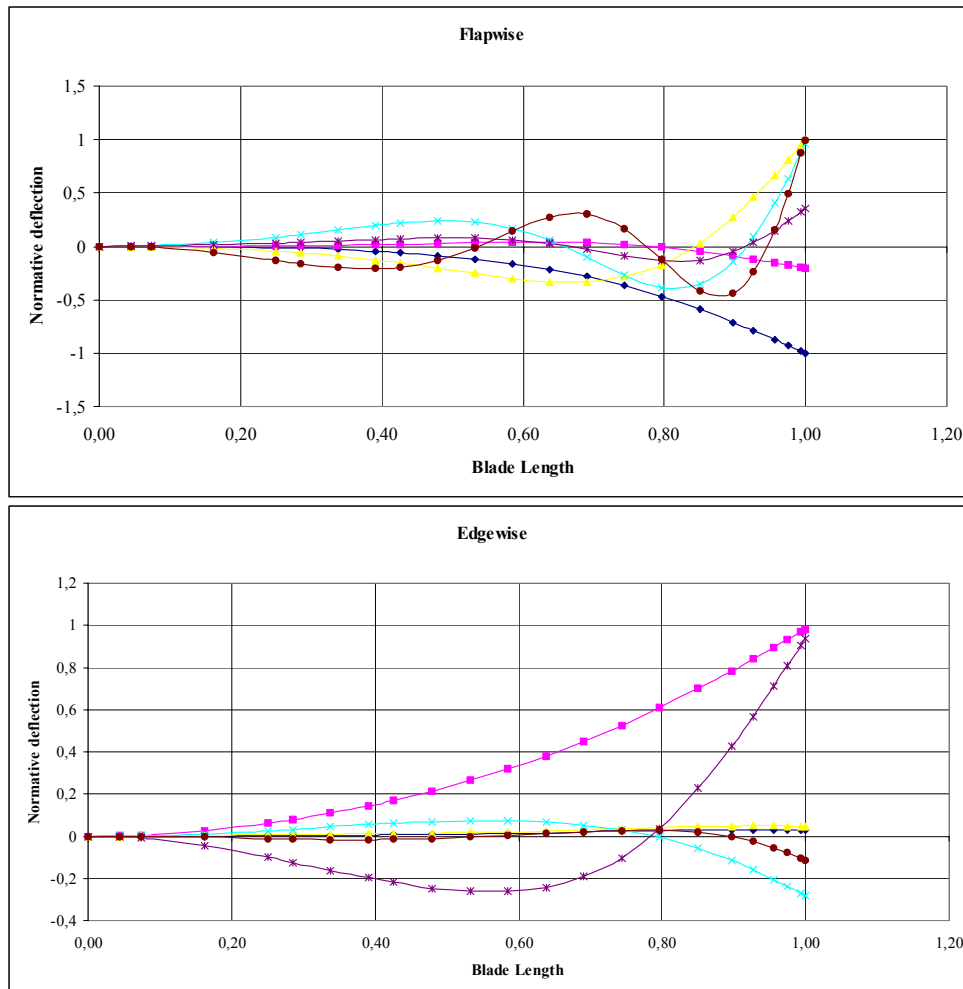


Figure 39. Calculated mode shapes, based on Stig Øyes Blade_EV1 program

The blade was separated into cross-sections. They were partly determined by picking out distances in the z-direction where the computer model had its highest amplitudes for the different mode shapes, cf. Figure 39 and Figure 40.

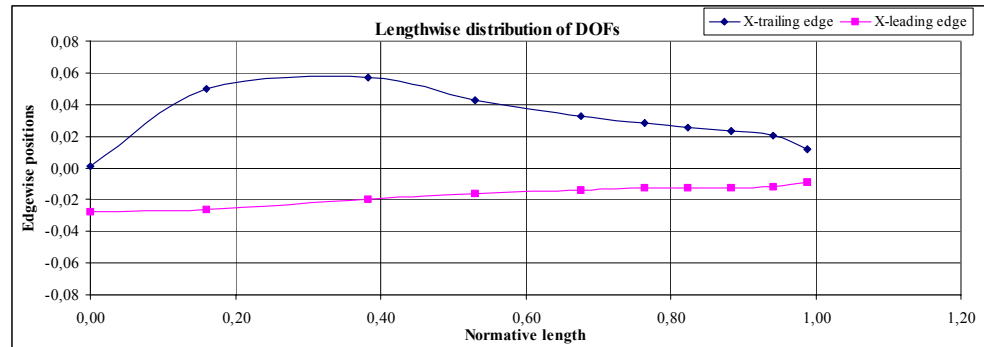


Figure 40. Lengthwise distribution of accelerometer positions on the LM blade

Experimental test set-up

The test was carried out at the test facilities at LM Glasfiber. The static test rig was used and the blade was mounted with the trailing edge pointing upwards. The accelerometers were mounted on the blade by use of steel plates as described in prior paragraphs. The blade was excited by use of the pendulum hammer.

Modal test

By measuring accelerations in two cross-sections simultaneously the test was carried out in five steps. The measurements were carried out from the root end towards the tip. For each cross section the blade was excited repeatedly.

Results

The results from the experimental test and the calculated results are compared. This shows good correlation between the experimental test and the model.

There is a difference between the deflection shapes of the model and the tested blade. It seems as if the blade in contrast to the model is stiffer. This is caused by differences in the properties of the blade and the model. There are also inaccuracies in the measurements. Finally there are differences in the joint between the blade and the test rig, and the boundaries of the model.

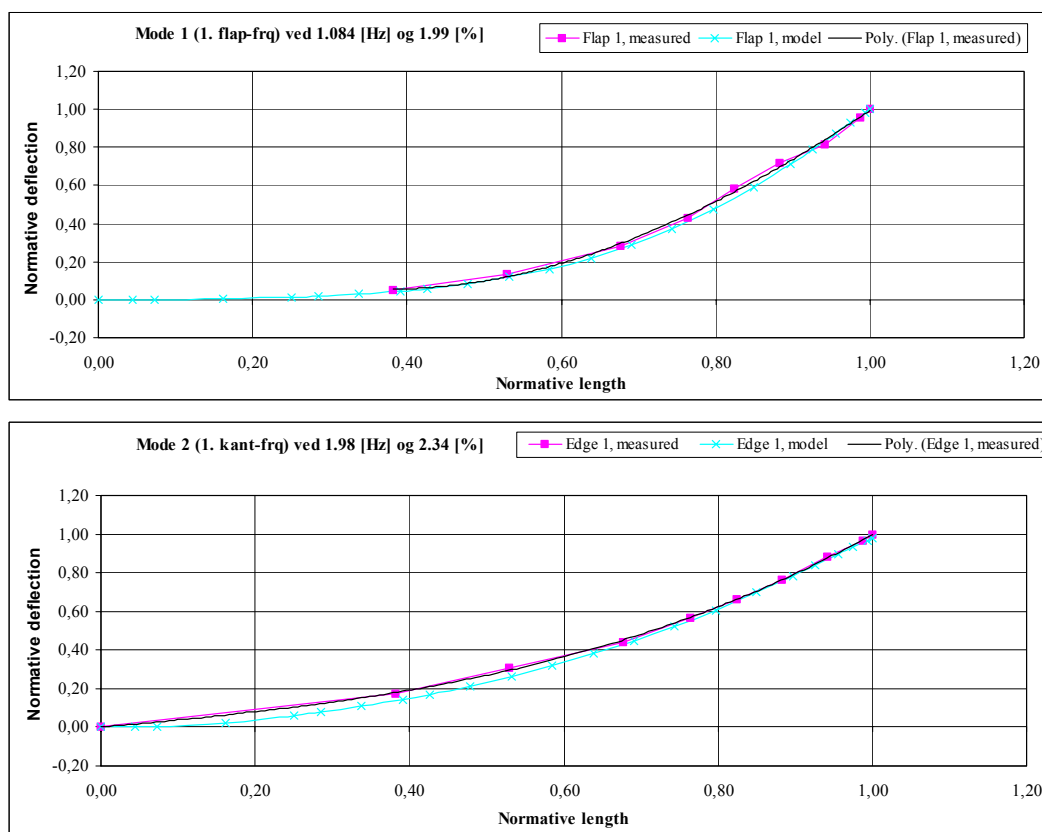


Figure 41. Comparison of computed model and experimental test, first flapwise mode and first edgewise mode

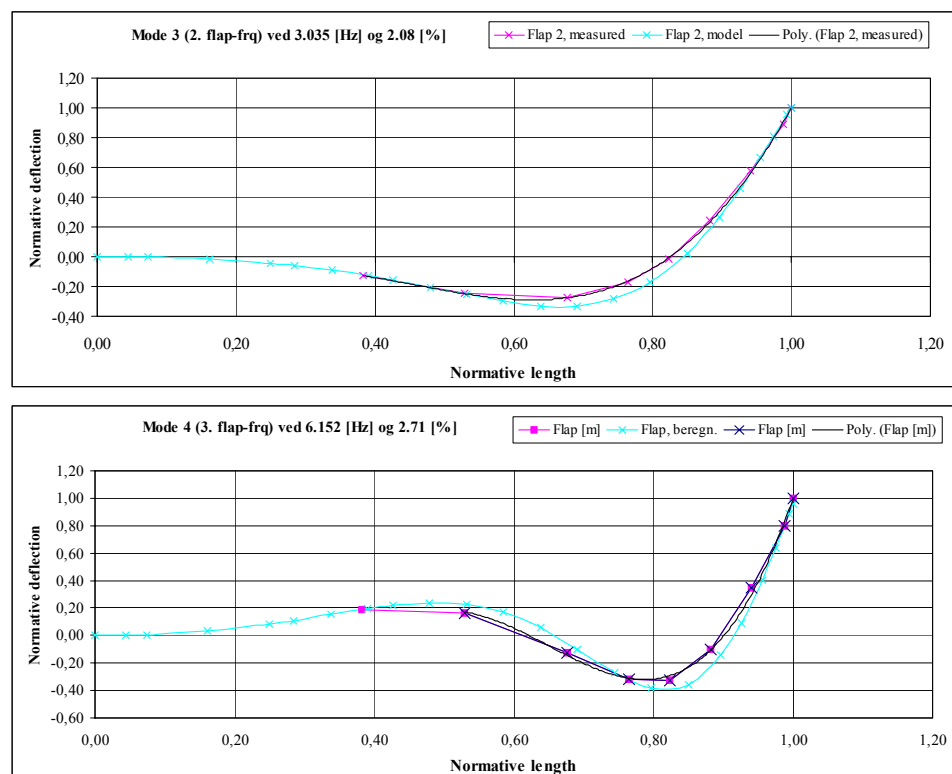


Figure 42. Comparison of computed model and experimental test, higher order modes

Table 10. Comparison of frequencies determined by use of Stig Øyes Blade_EVI program and frequencies determined by experimental test.

Mode	Model	Experimental results
#	[Hz]	[Hz]
1	1.04	1.08
2	2.13	1.98
3	3.01	3.04
4	6.43	6.15
5	7.05	6.60
6	11.56	10.48
7	-	12.32

The model does not calculate the torsional mode, mode 7 is the first torsion mode.

Conclusion

The test showed a good correlation between model and blade. The modal parameters, stiffness and mass, determined by the modal analysis can be used as input to the model.

9 Conclusion

In this project modal analysis proves that it can be used to determine dynamic characteristics of wind turbine blades. The most important properties like natural frequencies damping and the mode shapes can be determined experimentally by the modal analysis method.

The measurements have been performed on three different wind turbine blades. The length of the blades used during the experimental campaigns varies between 7.5 m and app.35.0 m. It is believed that the results also are valid for blades exceeding this size.

Different methods to measure the position and adjust the direction of the measuring points are discussed. Different equipment for mounting the accelerometers are investigated and the method using an angular steel plate was chosen. It was considered to be the most useable choice, when taking the time consumption during mounting and the necessary accuracy into account.

Different excitation techniques are tried during experimental campaigns. It was concluded that the pendulum hammer gives the highest accuracy in replication of the impact. In the project the pendulum hammer was improved, and a new pendulum hammer was manufactured.

Some measurement errors were investigated. The ability to repeat the measured results was investigated by repeated measurements on the same wind turbine blade. The measurements show that the results are very similar, when the measurements are repeated. It is concluded that the method is very reliable, with a high ability to repeat it self.

Furthermore the flexibility of the test set-up was investigated, by use of accelerometers mounted on the flexible adapter plate during the measurement campaign. It was demonstrated that the flexibility was depended on the natural frequency of the test rig and the fastening of the wind turbine blade to the test rig. Usually the measured frequencies are low and the system of fixture has a higher natural frequency. For that reason it is believed that the flexibility of the fixture normally is of minor importance.

One experimental campaign investigated the results obtained from a loaded and unloaded wind turbine blade. During this campaign the modal analysis were performed on a blade mounted in a horizontal and a vertical position respectively. The experimental campaign shows that the orientation of the blade has an effect on the measured results. There are minor deviations in the natural frequencies. The damping shows large deviation properly because of small quantities. In the mode shapes there are a good correlation, especially in the lower frequencies.

Finally the results obtained from modal analysis, carried out on a wind turbine blade are compared with results obtained from the Stig Øyes blade_EV1 program. Comparison between the results obtained with this two methods, shows a good correlation when comparing the frequency and mode shapes.

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Appendix

Equipment used during tests

Hardware	Type	PFV-no. / Serial-no.
Accelerometer	B&K type 4381V	1591
Accelerometer	B&K type 4381V	1592
Accelerometer	B&K type 4381V	1593
Accelerometer	B&K type 4381V	1594
Accelerometer	B&K type 4381V	1595
Accelerometer	B&K type 4381V	1596
Force transducer	B&K type 8201	1727594
Amplifier	B&K NEXUS	2226641
Amplifier	B&K NEXUS	2226642
Front-End	B&K type 2816	2143605
PC	HP Vectra VL	FR93912128
Software		
PULSE LabShop	Version 4.2	-
MEScope VES	Version 2.0	-

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Abstract (max. 2000 characters)

In this project modal analysis has been used to determine the natural frequencies, damping and the mode shapes for wind turbine blades.

Different methods to measure the position and adjust the direction of the measuring points are discussed. Different equipment for mounting the accelerometers are investigated and the most suitable are chosen.

Different excitation techniques are tried during experimental campaigns. After a discussion the pendulum hammer were chosen, and a new improved hammer was manufactured.

Some measurement errors are investigated. The ability to repeat the measured results is investigated by repeated measurement on the same wind turbine blade. Furthermore the flexibility of the test set-up is investigated, by use of accelerometers mounted on the flexible adapter plate during the measurement campaign. One experimental campaign investigated the results obtained from a loaded and unloaded wind turbine blade. During this campaign the modal analysis are performed on a blade mounted in a horizontal and a vertical position respectively.

Finally the results obtained from modal analysis carried out on a wind turbine blade are compared with results obtained from the Stig Øyes blade_EV1 program.

Descriptors INIS/EDB

ELASTICITY; MECHANICAL VIBRATIONS; NONDESTRUCTIVE TESTING; RESPONSE FUNCTIONS; STRUCTURAL MODELS; TURBINE BLADES; WIND TURBINES